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THE JOURNAL OF THE SOCIETY OF AUTOMOTIVE ENGINEERS

NOVEMBER 1922

PRODUCTION NUMBER

SOCIETY OF AUTOMOTIVE ENGINEERS INC.
29 WEST 39TH STREET NEW YORK

Plain Language *on* Easy Riding

IMPACT with a small bump means that the car springs will be compressed only slightly. And being compressed only slightly means that their recoil force will be very mild—just enough to raise the body back to normal. To oppose such mild force with any but a very light resistance would interfere with recovery to normal, and harsh, jerky riding would result.

And yet, impact with a larger bump means that the car springs will be compressed more fully, and then their recoil force will be more violent. A small braking resistance against this more violent recoil force would be useless because it would not be sufficient to prevent this violent force from tossing the car body and passengers.

And again, impact with a still larger bump means that the car springs will be compressed still more fully and then their recoil force will be still more violent. For this latter force we must therefore provide a still greater braking resistance if we are to prevent tossing of car body and passengers.

Watson Stabilators, for the first time in the history of recoil braking, oppose each varying mildness or violence of spring recoil with a resistance proportional to the recoil force. Their action is automatic, instantaneous and positive.

The H. H. Franklin Manufacturing Company have written a complete Bulletin covering their experience with Watson Stabilators. The following quotation from this Bulletin is significant:

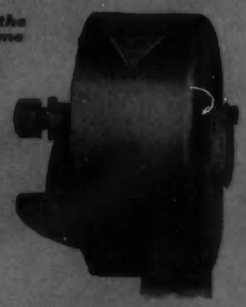
> "Here is a spring recoil device that Franklin Engineers finally can recommend, endorse and boost without reservation. *It is the first equipment of its kind that really does the trick.*"

And here's another one from this same Bulletin:

> "In the great mileage tests given at the Factory, no adjustments have been necessary. On one particular car driven 25,000 miles they are behaving as nicely as the day they were installed."

If you want to know what real-control of spring recoil means to the roadability of a motor car, you want to get acquainted with Stabilators.

Look for the Silver Name Plate



Exactly opposite to snubbing

In checking spring recoil, Stabilators work exactly opposite to the snubbing principle. Instead of checking with a jack at the tail end of the recoil movement, Stabilators get on the job at the very beginning of the movement and smoothly ease you back to normal. Results produced by the one method give no conception of those produced by the other. They are different to the point of absolute oppositeness.

John Warren Watson Company
Twenty-Fourth and Locust Streets
Philadelphia, Pa.

WATSON STABILATORS

CHANGE THE WHOLE NATURE OF YOUR CAR

THE JOURNAL OF THE SOCIETY OF AUTOMOTIVE ENGINEERS

Vol. XI November, 1922 No. 5

Chronicle and Comment

National Screw Thread Commission

GEORGE S. CASE has been appointed by Secretary of Commerce Hoover a member of the National Screw Thread Commission, in place of Edwin H. Ehrman whose resignation was accepted with regret. The Act of Congress that established the Commission provided that the Society should nominate two members thereof. Earle Buckingham, appointed by the Secretary of Commerce under nomination by the Society, is also serving as a member of the Commission.

Part 1, 1921 Transactions

COPIES of Part 1 of Volume XVI of the TRANSACTIONS are being distributed by parcel post this month to the members who have ordered them. The issue contains about 48 papers and reports, together with the discussion thereon, presented at the 1921 Annual Meeting of the Society and at various meetings of the Sections. The volume is constituted of over 800 printed pages. Less than 1500 members ordered copies of this Part of the TRANSACTIONS. There was no charge to the members for this volume in addition to payment of annual dues. This will likewise be the case with regard to the three next forthcoming Parts, namely those for the last half year of 1921 and for the two periods of 1922.

Gear Noise

GEAR noise persists as one of the industry's annoying problems in spite of continued study of gear mechanism over a long period of years. Basic tooth-form, pitch, pressure angle and addendum have been varied without material progress; non-metallic materials have been used with some success, but there is still room for improvement from the standpoint of decreased wear and noise. Two papers on gearing were presented at the Production Meeting of the Society in Detroit and both are printed in this issue of THE JOURNAL. K. L. Herrmann's paper directs attention to appreciable errors in form and spacing that are common in gear teeth cut by our present-day machine tools and methods. The observations given represent the findings of a searching investigation into the causes of gear noise by the Studebaker production organization. The paper by Messrs. Crain and Brodie records some rather unusual discoveries made by Packard production engineers who conclude that the mounting of gears is an important factor in the reduction of noise. Both papers deserve most careful study by automotive designing and production engineers.

The Production Meeting

PRIOR to the Production Meeting of the Society that was held in the General Motors Building at Detroit last month, one of the men who has been very prominent in the automotive industry for a number of years said

> I am quite willing to concede that I know something about production, and believe that I can originate ways and means to produce an article at as low cost as anyone else, yet I do not possess the ability to transmit that knowledge to an organization such as the Society of Automotive Engineers. I would be delighted if I could sit in as an interested listener at such a meeting as you are going to hold.
>
> I would also like awfully well to be able to make the trip through the various Detroit plants enumerated. It would be an education in itself.

Though unduly modest as to his ability to impart knowledge to others, this man foretold in a strikingly accurate manner what the Production Meeting proved to be. The taking up of production matters at a national meeting of the Society was a distinct success. It was a thing for which there was much need.

New Data Sheets

DATA sheets incorporating the Standards and Recommended Practices last approved by the Society, following the White Sulphur Springs meeting held in June, have been mailed to the members. These 44 sheets, dated August, 1922, present new or superseding matter. They are *for use* in connection with current and new designs. They are the result of able conscientious work by many members. They are not only worthwhile but an incomparable source of essential information for designers and their associates in the automotive field.

Instructions as to assembling the new sheets in the S.A.E. HANDBOOK accompany this last shipment of Standards information to the members. An important feature of the service is the readiness of the Society to furnish

without charge to members copies of sheets that are missing from but should be contained in their S.A.E. HANDBOOKS. A check-list of all live data sheets has been mailed them in this connection. Keep your S.A.E. HANDBOOK uptodate. Are you getting the benefit of the use of the S.A.E. HANDBOOK in your work? Are your employers, and in turn the users of your product getting the benefit of it?

Fuel Research Tests

AT a meeting of the steering committee held in the City of Washington at the Bureau of Standards on Sept. 20 and attended by representatives of the automotive and the petroleum industries and of the Bureau, the results already obtained from the cooperative fuel-research program were reviewed and examined critically. A vote taken on the advisability of continuing the tests during the winter was passed in the affirmative.

It is gratifying to note that only about one-half of the sum allotted by the industries for this work has been expended uptodate, and that the program originally provided for in the estimates is almost complete.

Automotive Engineer and Highway Engineer

THE automotive engineer needs data of rolling and air resistance rather than data of gasoline consumption from road tests, since the latter depend upon the individual characteristics of the vehicle used, the carbureter and the habits of the driver. Having the power factors, the automotive engineer will design the plant to furnish the power required.

From the standpoint of the highway engineer, a problem like the following is set:

A reduction in grade of 4 ft. will reduce the amount of energy required 8000 ft-lb. What will this be worth in the saving of gasoline and other operating expense for the traffic considered? Will the solution be for cars as built at present, or for cars using the roads 15 years from now? The mutual relation of the highway and the vehicle is expressed in the effect of low grades on changes in the engine design and the power requirements of the car. Inasmuch as the large expense of highway transportation is in the expense of operation of rolling stock rather than in fixed charges and maintenance of the road, the investigation of the costs of operation of traffic is particularly needed, it was felt at a Conference on Tractive Resistance of Roads held under the auspices of the Advisory Board on Highway Research of the National Research Council.

It is considered that a 5-per cent grade on a hard-surfaced road is not excessive for passenger cars and 2 to 3 per cent for large-capacity trucks. The fuel cost and the time required, as affected by the type and design of the surfaces and grades, are measurable by experimental methods, and the depreciation of the vehicle and other maintenance costs should be evaluated from a study of the statistics of motor vehicles obtained in actual operation.

Recording Car Road-Performance

ONE of the most important accomplishments in research is the perfection of instruments and appliances that makes it possible to observe and record conveniently and accurately just exactly what we want to know. Sometimes it is even harder to find out exactly what to record than to devise a means for recording it.

One of the best recent examples of a success in this line is furnished by the very ingenious combination instrument for recording almost all one needs to know concerning the performance of passenger cars on the road that was shown by W. S. James at the Summer Meeting. A complete description of this outfit will be published in THE JOURNAL in the near future.

Briefly, this recorder places on a road vehicle practically all the facilities of the usual dynamometer laboratory and several others besides. Moreover, the records are entirely automatic and yet in most of the measurements they have all the precision commonly obtained on the test stand. It should go far to settle the ancient quarrel between the exponents of laboratory tests and of road tests by putting the laboratory on the road.

The apparatus consists of a photographic recorder, which records the rate of fuel consumption, and another instrument with recording pens that gives on a single strip of paper a record of the following quantities:

(1) Engine speed
(2) Manifold depression
(3) Temperature of inlet water
(4) Temperature of outlet water
(5) Temperature of oil
(6) Temperature of the differential
(7) Temperature of air at the entrance to the carbureter
(8) Temperature of the atmospheric air
(9) Weight of air being used by the engine
(10) Acceleration of the car
(11) Air speed
(12) Apparent direction of the wind

Grade Transfer of Members

ON p. 127 of THE JOURNAL for August a comprehensive statement was given of the policy of the Society in the grading of applicants for membership. The Council may transfer a member from one grade to another, upon written request from him and the presentation of evidence that he is qualified for transfer to the other grade. The most usual cases are of course the applications for transfer from Junior or Associate to Member grade. There are no limitations of age for Associate grade. The Junior grade is open to those who are under 30 years of age and engaged in technical work. A Junior may on reaching the age of 26 be transferred to Member or Associate grade; upon reaching the age of 30 he may not retain Junior membership, but must be transferred to one of the grades mentioned. Upon transfer, he is required to pay the difference in initiation fee of the grades.

An Associateship is not as such *inferior* to a Membership, although the latter only carries voting and office-holding power in the Society. The fundamental difference is that of line of work or qualification. Associate grade is composed of those who are engaged in the automotive or related industries in responsible commercial, financial or manufacturing capacity. A principal feature is qualification to cooperate with automotive engineers.

Associates or Juniors who feel that they are entitled to Member grade should transmit written application therefor to the office of the Society, stating what work they have done since the time of their election that would in their opinion justify the transfer, and giving the names of three members as references who know of this work. Such applications are given careful attention by a committee appointed by the Council, the latter in turn giving due consideration to the written records and the committee recommendations.

The Production Meeting

THE unusual degree of success attained by the Production Meeting in Detroit, Oct. 26 and 27, is attested best by the size and character of the attendance. Close to 600 persons registered at the meetings, factory visits and the Dinner. They came from all corners of the Country, some from Minnesota and Wisconsin, others from Kentucky, Iowa and Massachusetts. Production managers, factory superintendents, tool supervisors, time-study men, chief inspectors and others of similar high rank constituted the major portion of the attendance. It is difficult to imagine how a meeting could have been more representative of a single branch of an industry.

The Production Dinner attracted close to 400 members and guests. The two professional sessions included audiences that averaged 350 persons and at times passed the 400 mark. This is a record for professional meetings of the Society, either sectional or national. As tangible approval of the Society's purpose to enter the production field actively, these figures and the attentiveness and enthusiasm of those present could hardly be surpassed. The Production Meeting is established as a Society institution without question.

Morning Sessions

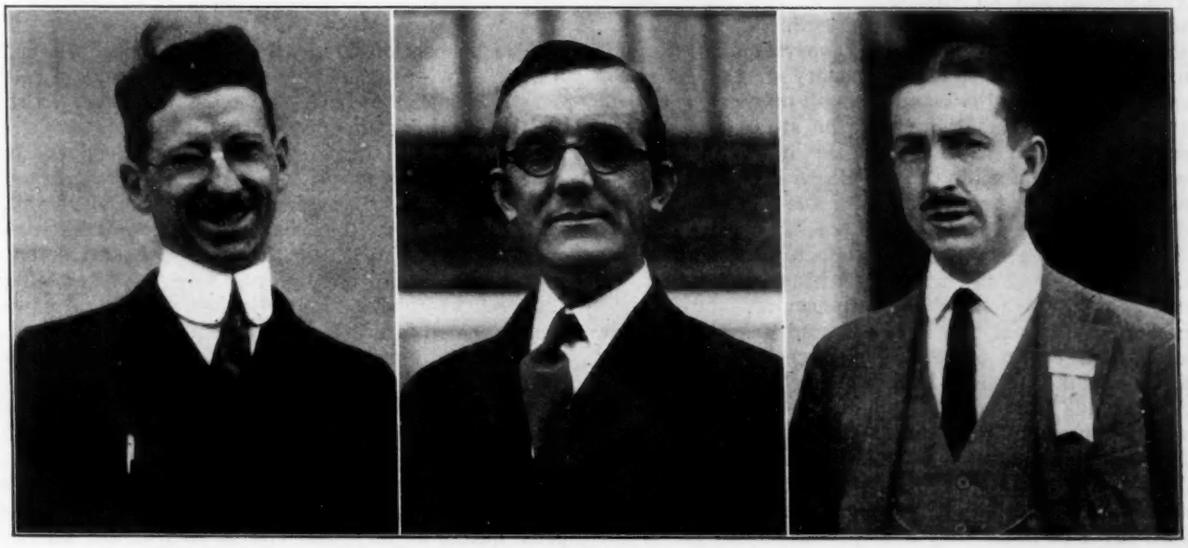
The Thursday morning session was convened at 9:43. H. W. Alden, as presiding officer, observed that the builders of the Society Constitution had proceeded wisely in providing for production a place in our activities equal to that of design. The day of experiment and design has of course not passed; these are more important than ever before. But production problems must have more and more attention. The engineer must know about production. The production man must know something of engineering.

E. Karl Wennerlund, who presented the first paper of this session, prefaced his able oral outline of it by giving a sketch of the stages of industrial development which preceded the present status of wage incentives; beginning at the time when foremen set piece prices based on their judgment. There was little method of checking quantities of articles produced until a few years ago. Even to-day probably 50 per cent of industry does not have this quantity check. The coupon system of checking production became impractical in factories in which thousands of operations were going on. To-day the system of numbering lots serially is generally used. An effort is being made to depart from this practice which is too unwieldy in large production in which several thousand men are employed; it being necessary commonly to check 15,000 tickets daily.

Mr. Wennerlund believes that the group-bonus wage-incentive plan, which is described in his paper printed in this issue of THE JOURNAL, is the best system for application in the automotive industry; having in mind the use of the progressive system of manufacture, in which, generally speaking, the work in process does not touch the floor. He is of the opinion that it combines the merits of the piece rate so far as incentive is concerned and of the day wage in respect to flexibility.

The paper by F. A. Mance, on the Control of Operating Tool and Supply Cost, which is printed in this issue of THE JOURNAL, dealt briefly with means taken to regulate the number of tools and amounts of supplies used in car-building factories. Reference was had to a system that has been in effect in the Studebaker plant for 4 years, with a resultant decreased cost of renewals of perishable tools. In connection with studies of the life of perishable tools, some data with regard to drills were given.

In a paper by James A. Ford on Processing Spline Shafts by a New Method, the tools used in overcoming the difficulty in hobbing to exact dimensions were described; these being a die constructed in a manner similar to that followed in the fabrication of automatic threading-dies, the cutters being in the position that the



K. L. Herrmann Thomas J. Litle, Jr. K. K. Hoagg

THREE MEN WHO WERE LARGELY RESPONSIBLE FOR THE SUCCESS OF THE MEETING

die chasers ordinarily occupy; and a cam ring of strong construction. The methods employed to obviate warpage during the hardening process and in re-centering shafts were also set forth. The paper appears on another page of this issue.

The paper by P. E. Haglund and I. B. Scofield presented the fundamental reasoning underlying the creation and operation of the Ford River Rouge Plant. Basic raw materials, coal, iron ore and limestone, are brought to this great industrial unit at minimum cost, conveyed mechanically to coke-oven, blast-furnace and foundry, all by-products being utilized at each step and industrial waste practically eliminated. Coke, combustible gases, tar, chemicals for fertilizer and benzol are used in plant operation or sold commercially. Blast-furnace gas is burned in the powerhouse. Slag will be used eventually in cement manufacture. Molten blast-furnace iron is conveyed direct to the foundry, mixed with cupola iron and poured, so that the heat generally wasted in casting pigs is conserved. Sand from the foundry molds returns automatically to the molders while still possessed of some of its heat from the previous casting.

Molds are assembled, cores made and set, castings poured and cooled, all on a conveyor system. Cylinder blocks are conveyed mechanically through shake-out, cleaning and inspection departments direct to a compact machining department where they are machined without a pause or return to the floor and passed out to the finished-stock loading platform. Truly, here we find American production genius at its height. This paper, fully illustrated, will appear in the December issue of THE JOURNAL.

President Bachman presided at the Friday morning session. H. P. Harrison presented the paper on some phases of construction of the Franklin automobile, this being entitled Some Unique Features of Automobile Construction, which is printed on p. 398. The production of air-cooled engine cylinders, crankshafts with case-hardened bearings, forged aluminum-alloy connecting-rods and hot-swaged rear-axle drive-shafts were discussed in turn.

L. C. Hill presented the contribution by H. J. Crain and J. Brodie, of the Packard company, under the title, Experience Notes from a Production Notebook. In this interesting narrative brief descriptions were given of machinery devised for the purpose of running-in brake-band assemblies and of tools for centering and driving pistons during external-grinding operations. In addition, points involved in the discovery of the causes of obscure noises in engines and gear oil-pumps were given, and in particular an account of the study the Packard company has been making of gear noises in general, especially in connection with transmissions and rear axles. The paper is printed in this issue of THE JOURNAL.

Many matters relating to gears, including noises, tooth spacing, variations in finished product, index errors and tooth-forms, as well as cutting machines, hardening and grinding, were treated by K. L. Herrmann in his paper, Some Causes of Gear-Tooth Errors and Their Detection, which is printed on p. 391. His purpose was to show that production variables have a much greater influence on gear sounds than pressure-angles, steel or tooth-forms. In this connection he sought the assistance of gear-cutting tool and machine designers. He urged strongly the importance of this, pointing to the large apparently unnecessary expense in the production of gears, and also expense involved after they are "produced."

It was evident not only that Mr. Herrmann had a comprehensive grasp of his subject, but that he had taken great pains to make clear essential matters involved in improving the conditions of the quantity production of gears. Supplementing his paper, he projected some views of gears in operation to illustrate tooth-spacing errors, interferences and other undue variations. It is earnestly hoped that Mr. Herrmann's efforts will stimulate incisive and clarifying discussion at future meetings of the Society and the Sections.

Two papers on the Friday program were the occasion of a most interesting incident in that two production authorities reached identical conclusions independently on the important matter of selecting machine-tools. A. J. Baker and R. K. Mitchell, of the Willys-Overland and the Maxwell plants, respectively, read the papers in question. They both believed that the efficiency of the specially designed and highly productive machine-tool has been greatly exaggerated and predicted a movement toward wider use of standard machine-tools that can be adapted to various classes of work by the employment of simple and inexpensive fixtures and attachments. Mr. Mitchell said that the most forceful argument against special machine-tools is the unstable design and continuous development of the automotive parts themselves. He recommended a more general use of the three fundamentals of jig and fixture work: the clamp, the V-block and the angle-plate.

Mr. Baker's conclusions were as follows:

(1) There is a surplusage of machine-tool equipment of the standard types, both actual and potential
(2) Machine-tool builders are devoting their thought to high-production single-purpose machines of standardized types
(3) The craze for special machinery is passing
(4) Special machinery will not always stand a financial comparison with standard machinery
(5) We are not, as an industry, facing our responsibilities in the matter of training operative help for tool and die help
(6) We cannot disregard the inventory value of existing equipment and the loss that will be shown on our balance-sheet by its conversion from productive machinery to excess machinery for sale when considering new equipment
(7) The only good reason for installing new machinery, old machinery or any machinery, apart from those cases where higher quality is demanded, is to reduce cost

FACTORY VISITS

The inspection trips through representative Detroit factories were a much appreciated feature of the meeting. Over 250 production men participated in the visit to the Ford River Rouge plant on Thursday afternoon. The party was taken through the by-products building where the reclamation of tar, benzol and ammonium-sulphate from the coking process was observed. The coke-ovens, blast-furnaces, foundry and machine-shops were all seen in operation. Sections of the tractor assembly, stamping and body plants were inspected. The visit was concluded with a trip through the central powerplant which has an interesting boiler equipment that utilizes all manner of by-products in its novel combustion system. The members were impressed by the extensive use of mechanical conveyors in all departments, the elimination of waste material and labor and the relatively small number of men required to operate this huge project.

Three simultaneous visits were made Friday afternoon to the Packard, the Dodge and the Cadillac factories. Approximately 60 members visited each of the plants. The party visiting the Cadillac factory was shown through the engine, clutch, transmission and chassis-assembly de-

A. B. C. Hardy
A. J. Baker
H. P. Harrison
I. B. Scofield
E. K. Wennerlund
R. K. Mitchell
H. W. Alden
P. E. Haglund
B. B. Bachman

AUTHORS AND PROMINENT SPEAKERS AT THE PRODUCTION MEETING

partments. Cadillac assembly methods are unusual in that the mechanical or gravity conveyor is not used extensively. Attention is given to the completion of a number of bench assemblies that are taken later direct to one general assembly group and distributed to the men. The men are moved rather than the work in making the progressive major assemblies. The party inspected the gear-cutting department and saw a group of new gear-grinding machines in operation. A trip through the heat-treating plant and an inspection of the fender enamelling system was included in the Cadillac visit.

The members who visited the Dodge factory were impressed particularly by the number of interesting stampings produced in the press-room. The body and fender enamelling system, which is the largest of its kind, was explained to the visitors. The members were shown parts of the engine-assembly line, the chassis-assembly and some of the more interesting machine operations. The efficient use of floor space to increase the productivity per square foot was one of the outstanding impressions left by the Dodge visit. It was also noticed that careful preparation of the steel surfaces before enamelling is largely responsible for the excellent finish given the enamelled body and parts.

The Packard visitors were shown through all of the major departments of the passenger-car division with the exception of the small-parts machine shops. The foundry, forge and heat-treating plants were inspected and the party taken through the cylinder and crankcase machining departments. Passing through the engine block-test room, the members were taken down the long conveyor line on which the chassis assembly is made. A group of men especially interested in truck construction were taken through the Packard truck shops.

The production staffs of the factories visited deserve special recognition for their hospitality and the efficient manner in which they conducted the visits. Inquiries were always answered with specific information, the unconventional practices of each factory were explained unselfishly and lack of time alone prevented the most scrutinizing type of inspection of each factory.

The Dinner

It is not too much to say that the Society never held a more interesting event, or one more important to the automotive industry, than the Production Dinner. President Bachman, in introducing Harold H. Emmons, the toastmaster, iterated emphatically that there is no essential difference in the work of those engaged in manufacturing and of those whose vocation is research or design. Moreover, that the man who feels that he in his own particular work is self-sufficient and not intimately connected with others in selling and keeping sold the articles on which our industry is founded, is sadly lacking in vision.

Pierre S. duPont, president of the General Motors Corporation, held the diners intently with very felicitous and forceful remarks. He said in part:

> If we work together in the industry, its problems will be easily solved. If we are not fearful of its problems, they will be still more easily solved.
>
> We are fortunate to be engaged in an industry that is an infant. The world is before us now; there is not only the realm of our own industry and its many ramifications and possibilities, but thousands of uses for the automobile that we do not picture to-day. We have only scratched the industry. This is the only country in the world that uses the automobile, practically. Yet the bulk of the population is elsewhere, with its own problems, peculiar if you will, but all to be solved; before you to be solved.

A. B. C. Hardy, president and general manager of the Olds Motor Works, was the principal speaker of the evening, his topic being "The Production Man's Place in Our Industry." Mr. Emmons said that Mr. Hardy presented the best analysis of this subject ever made.

Mr. Hardy pointed out that the production or manufacturing division of an automobile plant occupies from 80 to 90 per cent of the land acreage and from 70 to 80 per cent of the floor space involved, and directs the work of from 5 to 10 or even 15 times as many people as are required by all the other divisions of the plant combined. He listed in a suggestive way the things the production man must know, and spoke of how he must be schooled. He said, among other things

> No one accomplishes anything alone in our complicated fast-moving industry.
>
> The production man, if he be a wise man, is always on the lookout in his own plant, and also takes a side-look into other plants.
>
> The production man and his department heads are the means of conveying to the workers the standards and character of the company. And the workers judge the company accordingly.
>
> The real bosses of all our jobs are the motor-using public.
>
> I have seen some poor designing turned into fairly good cars by production work and refining.

By direction of Toastmaster Emmons Past-President Kettering gave the benediction. The benediction included the following remarks:

> When we engineers design a thing, and the production man starts to make it and puts it together, it is never what we designed.
>
> The time for the production engineer to "sit in" is before the thing is designed. The industry has spent hundreds of millions of dollars foolishly by not calling in the production man when the design was in process.
>
> The two closest fellows in any organization should be the production man and the engineer. A third man, who has never been there before, should come into the picture—the accounting man.
>
> This meeting marks an epoch in the history of the automobile engineer.

Committeemen and Co-Workers

The story of the Production Meeting would not be complete without mention of those whose efforts during the past three months contributed to its success. K. L. Herrmann and T. J. Litle gave much of their personal time in the solicitation and selection of the valuable papers. K. K. Hoagg not only arranged for the several factory visits but secured the transportation equipment and directed the efficient loading and dispatch of the vehicles. W. C. Keys, and the conscientious members of the Reception Committee, W. R. Strickland, Frank Ruoff, C. H. Brennan, A. L. Kimball and J. W. Spray, deserve thanks for undertaking and carrying out many detail responsibilities in a manner praised by all. E. F. Roberts, A. U. Widman, H. A. Coffin, George E. Goddard and C. E. Sorensen were the genial hosts who opened the doors of the factories to the members. The authors and speakers deserve much commendation, although they were amply repaid by the enthusiastic reception accorded their respective contributions. Acknowledgment is made to the General Motors Building Corporation for the provision of quarters for the meeting which were excellent in accommodations and facilities. The Detroit Section of the Society is thanked for the cooperation of its officers and

(*Concluded on p.* 408)

Some Causes of Gear-Tooth Errors and Their Detection

By K. L. Herrmann[1]

Detroit Production Meeting Paper *Illustrated with* Charts and Drawings

THE different gear noises are classified under the names of knock, rattle, growl, hum and sing, and these are discussed at some length, examples of defects that cause noise being given and a device for checking tooth spacing being illustrated and described. An instrument for analyzing tooth-forms that produce these different noises is illustrated and described.

Causes of the errors in gears may be in the hardening process, in the cutting machines or in the cutters. A hobbing machine is used as an example and its possibilities for error are commented upon. Tooth-forms are illustrated and treated briefly, and the hardening of gears and the grinding of gear-tooth forms are given similar attention.

MOTOR-CAR production-men, as a rule, do not lay claim to being specialists in all the various arts and sciences that enter into the finished motor-car; so, in the matter of gears, we do not pretend to know all the details that enter into their design and production. Very few of us are familiar with the mathematics relative to the involute gear-tooth forms that the engineering fraternity stresses considerably in connection with gear noise. We are much more familiar with the noises of the finished product, which are recognized

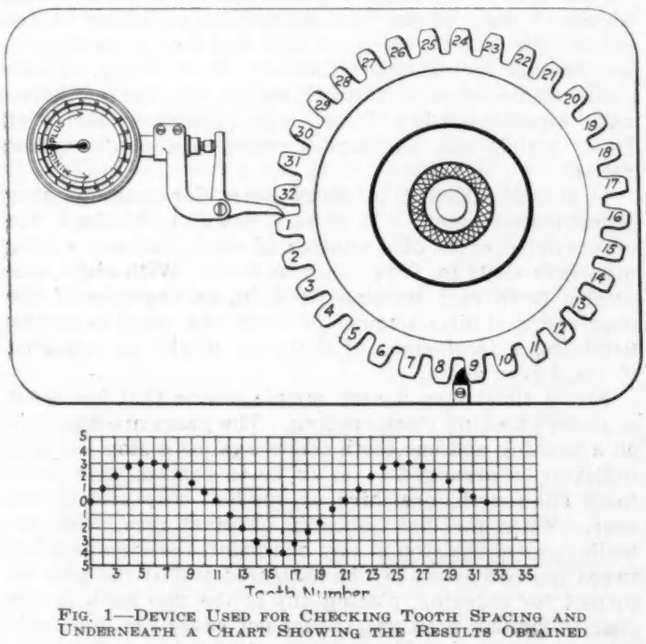


Fig. 1—Device Used for Checking Tooth Spacing and Underneath a Chart Showing the Results Obtained

as knock, rattle, growl, hum and sing. When these occur, naturally those men who have made a life study of gear subjects are called in, and they make recommendations.

As a rule our gear experts offer widely varying remedies for the same cause. If one is using a 20-deg. pressure-angle, we can advise a 14½-deg. angle and have a large number of supporters in both the engineering and the production fields. If both 20 and 14½-deg. angles have been tried, we can easily advise a stub tooth or full length. If all six combinations have been tried, it is easy to recommend topping-off or no topping-off; and, if this does not work, we recommend a different *steel*, *cutting machine* or *cutter*. While all of this goes on, sufficient time has elapsed to start all over, with the first recommendation. Of all of these recommendations we find many past and present supporters of national reputation.

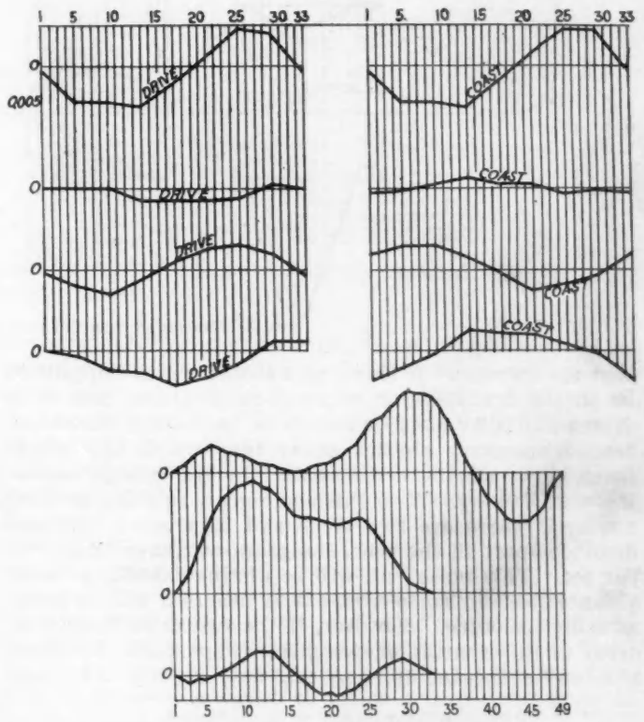


Fig. 2—Chart of the Variations in a Finished Gear

It is the purpose of this paper to show that production variables have a much greater influence on gear sounds than changing pressure-angles, steel or tooth-form details; also, by showing the errors present, to obtain definite help from the gear-cutting tool and the machine designer. We will confine this discussion, for the time being, to the transmission, which is the simplest type of gearing used in the motor car.

Gear Noises

We have already referred to the various kinds of gear noise. The first is a knock that might be caused by a nicked tooth or a single tooth of a pair of running gears

[1] M.S.A.E.—Engineer, Studebaker Corporation, Detroit.

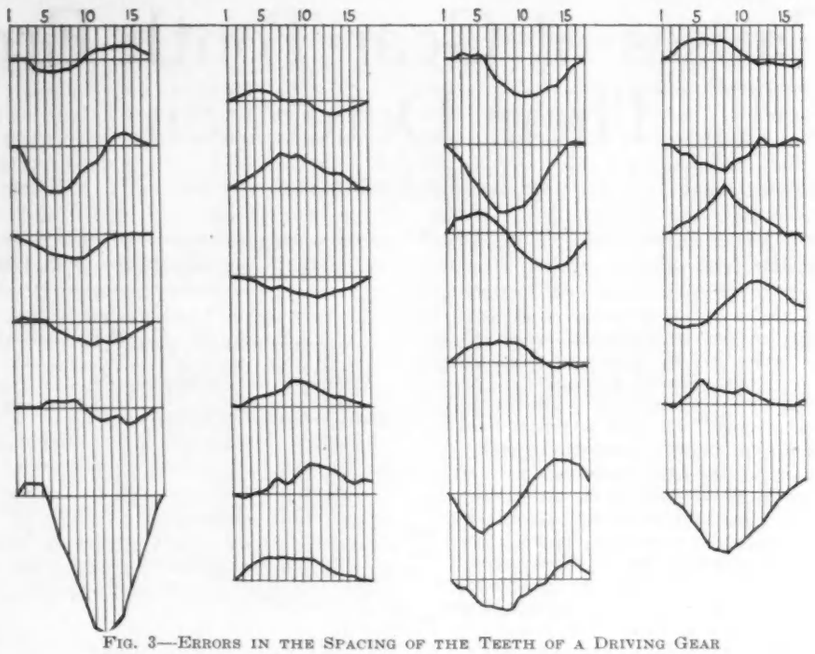


FIG. 3—ERRORS IN THE SPACING OF THE TEETH OF A DRIVING GEAR

that are in mesh. If this single tooth or nick happens to be in the transmission constant-mesh pinion and it is driven at 1000 r.p.m., there will be 16 distinct blows per sec. Elementary physics shows that, up to this point, these blows can be distinguished by the human ear as individual blows. With this same gear, rotating at 1000 r.p.m., if there are two nicks and they are a uniform distance apart in the gear, the gears will have 32 knocks per sec. This noise still will be distinguishable as individual knocks by some ears, but by others it will be noted as a distinct tune. However, if the two nicks happen to occur in this gear an uneven distance apart and the transmission is speeded-up, one can note readily the place where the gear noise changes from an individual knock into a sound that is usually designated as a rattle.

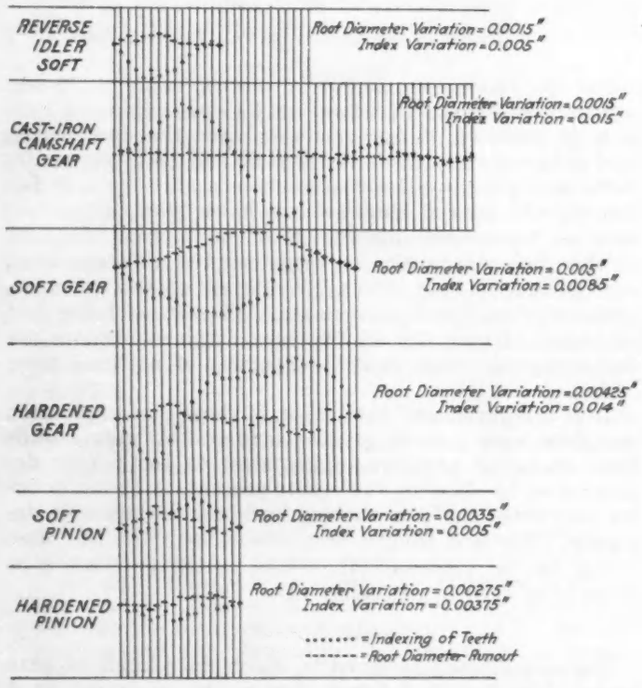


FIG. 4—CHART OBTAINED FROM A NUMBER OF GEARS AND PINIONS NONE OF WHICH HAS AN INDEX ERROR OF MORE THAN 0.005 IN.

It is evident that influences similar to the nicks referred to above can be produced by inaccurate conditions in the gears. For example, should the tooth spacing in the transmission drive-pinion mentioned be such that the driven gear, instead of rotating at a uniform speed, is forced to increase and decrease its speed at every revolution of the pinion, very similar sounds will occur. If on one side of the driving pinion the teeth are 0.015 in. ahead of their proper position on the periphery of the pinion, the driven gear must gain and lose a corresponding amount in its steady motion. In so doing, it may strike on the back of its tooth rather than on the driven side, especially when idling, and produce a series of blows which still further increases and confuses the rattle.

It is not sufficient to check gears for spacing error from tooth to tooth. It is very desirable to check the accumulated error of a number of teeth, because a gear may vary 0.001 in. from tooth to tooth. With eight successive teeth each gaining 0.001 in. on the side of the gear and a similar number of teeth that may be losing 0.001 in., a total error of 0.016 in. might be imparted to the driven gears.

Fig. 1 illustrates a very simple device that has been used for checking tooth spacing. The gears are mounted on a bushing and one tooth comes against a stop. A dial indicator is arranged so as to be in contact with some tooth one-fourth, one-third or one-half way around the gear. When the dial indicator is set at zero, with the tooth against the stop at any one point, the distances between the two points can be measured and, if the gear be correct for indexing, placing any of the two teeth in the gear in similar positions should not cause the dial indicator to vary, especially if the gear runs true.

When the gear is first put on the indicating apparatus, the dial indicator is set at zero. We then put a mark at zero on the chart in Fig. 1 for tooth No. 1. The next step is to index the gear around one tooth. Any reading obtained is marked above the tooth number in the vertical line. We next index the gear around to tooth No. 3 and again mark the dial-indicator reading

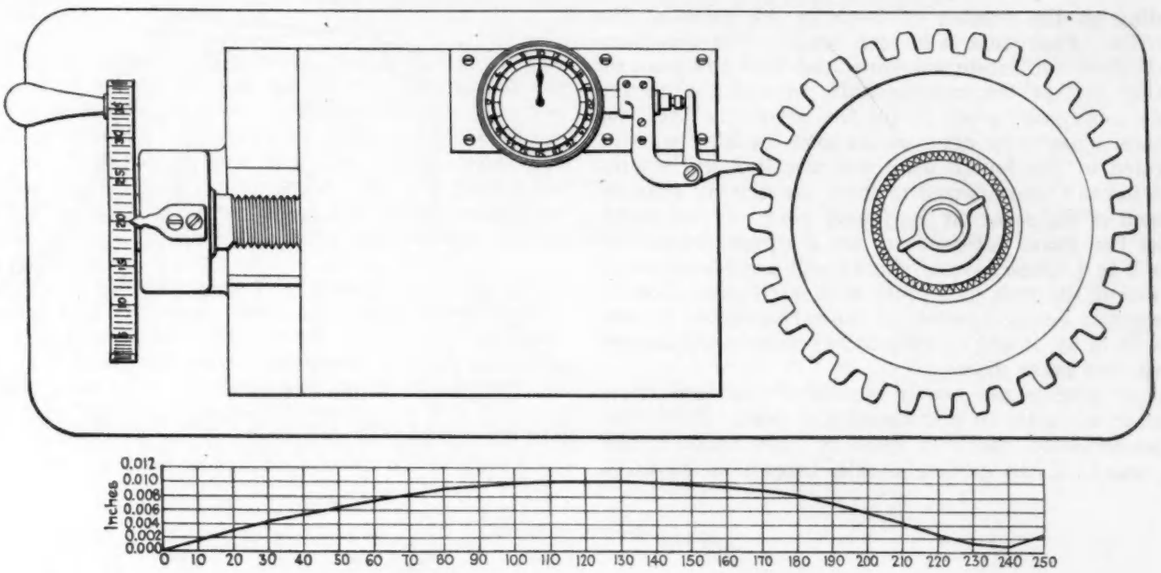


FIG. 5—INSTRUMENT FOR OBTAINING AN ANALYSIS OF TOOTH-FORMS PRODUCING NOISE AND A CHART SHOWING THE OUTLINE OF A PARTICULAR TOOTH

opposite the number of thousandths of an inch that it may show. The gear is then indexed to teeth Nos. 5, 6, etc., until all the teeth on the gear have been indexed.

For the purpose of record, we now have a chart showing the accumulated variables. It will be seen from Fig. 1 that at no point is the spacing variable as great as 0.001 in. between any two teeth, but it can be in error a total of 0.008 in. or more when the error between the several teeth has accumulated. A better visual demonstration of this condition occurring in gears is made by means of a gear-tooth-form projector. When a gear-tooth form is projected upon a screen by this device, it will be noted that the magnification of the shadow is 100 to 1 and that, for every inch on the screen there is at least a 0.001-in. error somewhere in the gear. The shadow on the screen also shows the variation in uniform movement of the driven gear due to an error of this kind; that is, the driven gear, instead of having its tooth in the position of the outline on the screen, has been forced to advance a number of thousandths of an inch. It will be noticed further that this advance and retardation does not occur uniformly; that is, the advance may be confined to a very small number of teeth, remain there for a certain length of time and then be retarded slowly. A gear in this condition will give a rattle very similar to that which might be produced by unequal spacing. This condition can be studied best by charting it as described.

A number of variations as they occur in the finished gear are shown. Fig. 2 shows, first, a rapid drop, then a flat portion showing no change, then a rapid rise, another flat and then a return to the starting point. These curves require very little explanation.

It requires considerable imagination to determine just what happens when a driven gear such as those described is meshed with a driving gear having teeth misplaced as shown in Fig. 3; yet this occurs in daily practice. Fig. 4 shows a number of gears and pinions none of which has an index error of more than 0.005 in.

In addition to the two gears having a rather irregular action between themselves, their increasing and decreasing movement is carried on to the countershaft and the idler, which often have similar defects in themselves. Accumulating errors in gears often cause the fourth gear in a train to be as much as 0.025 in. away from its correct position and this change occurs in varying amounts

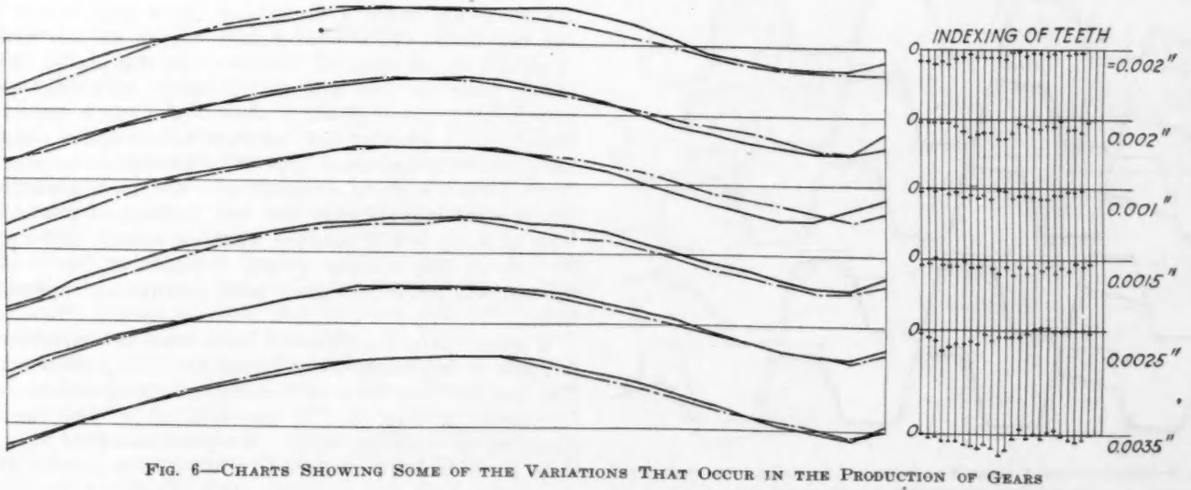


FIG. 6—CHARTS SHOWING SOME OF THE VARIATIONS THAT OCCUR IN THE PRODUCTION OF GEARS

depending on the number of teeth in the gears of the gear train. From this it is seen readily that the countershaft does not rotate smoothly and that the gear on the other end of the countershaft, in addition to the variable movement given it by the errors in the first two gears, imparts its error to the gear meshing with it. The action of the fourth gear will also lack uniformity in addition to that imparted to it by the driving gear, to the extent of the error on the fourth gear. If the ratios between the gears referred to are a direct proportion such as 2 to 1, these several shocks will occur uniformly, the error of the first uniformly with each revolution of the respective gears; however, if the ratios should be odd, such as 16 to 31, it will be difficult to estimate the change of speed that takes places.

Hum or sing is not nearly so difficult to analyze as the matter of rattle in a transmission gear. With this same transmission run at a speed of 1000 r.p.m. at the pinion shaft, if the pinion should happen to have 16

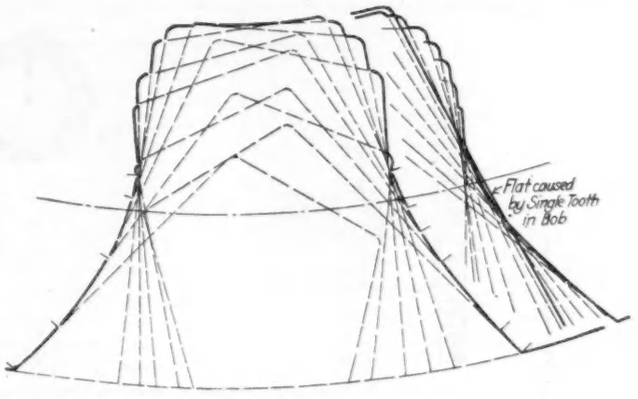


FIG. 9—A SINGLE HIGH TOOTH IN A HOB WILL PRODUCE A WIDER FLAT IN THE GEAR THAN IF ALL THE TEETH ARE OF UNIFORM HEIGHT

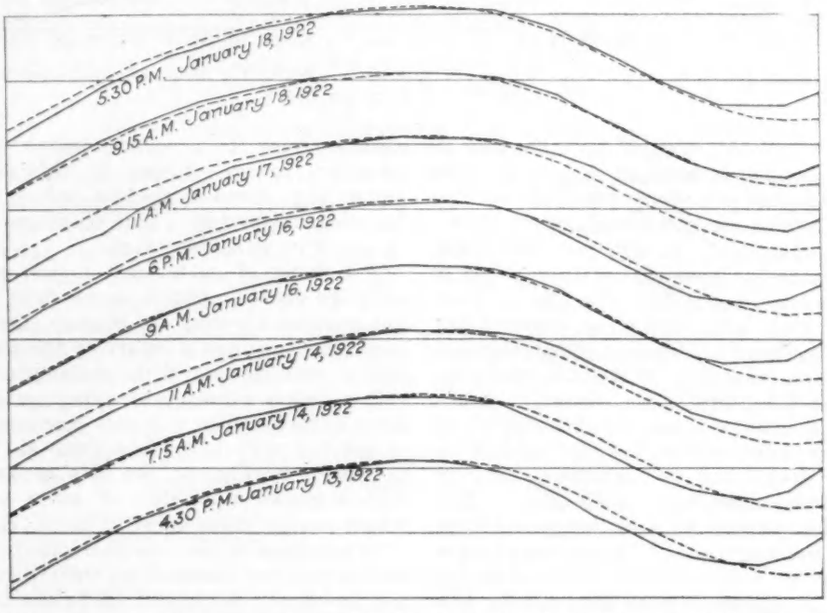

5.30 P.M. January 18, 1922
9.15 A.M. January 18, 1922
11 A.M. January 17, 1922
6 P.M. January 16, 1922
9 A.M. January 16, 1922
11 A.M. January 14, 1922
7.15 A.M. January 14, 1922
4.30 P.M. January 13, 1922

FIG. 7—VARIATIONS OCCUR IN THE PRODUCTION FROM THE SAME HOB ON DIFFERENT DAYS AND ALSO AT DIFFERENT HOURS OF ANY ONE DAY

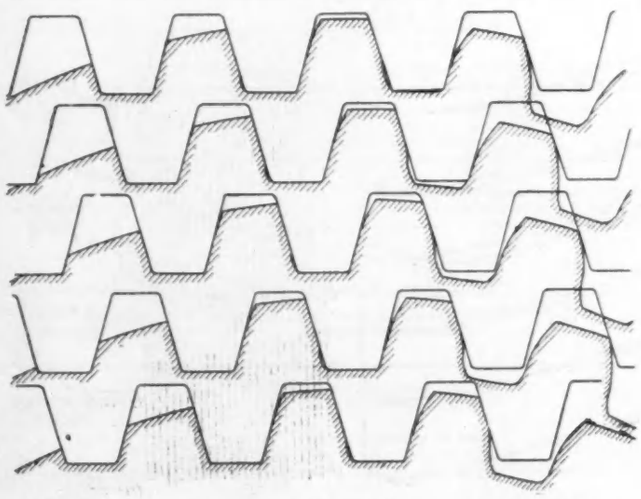


FIG. 8—OUTLINE OF A GEAR PRODUCED BY THE HOBBING METHOD

teeth, it will be found that 250 teeth per sec. go into and out of mesh. If the teeth are not correct in shape and the gears are under a slight load, there may be 250 blows per sec. under certain conditions, which we are told corresponds to the tone of middle C on the piano. Should fewer teeth go into and out of mesh, and this may be caused by a slower speed, a much lower pitch can be produced. In a similar way, because of the speed reduction that occurs in the usual type of transmission between the drive pinion and its countershaft, the tone produced by the reverse idler is very low and, instead of producing a hum or sing, it will produce what we usually call a growl. Errors in the sliding gears, because of their higher speeds, will produce higher pitch growls and approach a hum.

A great many instruments have been developed for the purpose of analyzing tooth-forms producing these sounds. The one that we have worked out and have chosen to use is shown in Fig. 5. It consists of a dial indicator mounted on a guided slide. We place the gear in a definite position with respect to the indicator, start at the

point of the tooth and set the indicator at zero. The slide is then moved toward the gear 0.010 in. and the indicator-reading marked on the chart shown in the lower portion of Fig. 5. The slide is then moved 0.010 in. more, the reading is marked again, and this is continued until the bottom of the tooth is reached. By taking the gear that has just been charted off the bushing and placing another gear in its place, other tooth-forms will be compared with the first.

The charts reproduced in Fig. 6 show some variations occurring in production, these particular charts having been made from gears with which special care had been taken to secure good gears. This was done by men who were not previously familiar with the method that was going to be used for the gear inspection. Fig. 7 shows some curves that occur in production from the same hob, and also curves made at different hours of the day.

Cutting Machines

The causes of the errors referred to are various. Some of them occur in hardening, some in the cutting machines and some in the cutters. We have found these errors in all of the types of machine that we have used. For the purpose of this discussion we are selecting a hobbing machine.

The hob is a generating tool that produces a gear such as is shown in Fig. 8. It will be seen readily that if all tooth-heights of the hob are the same, each hob-tooth

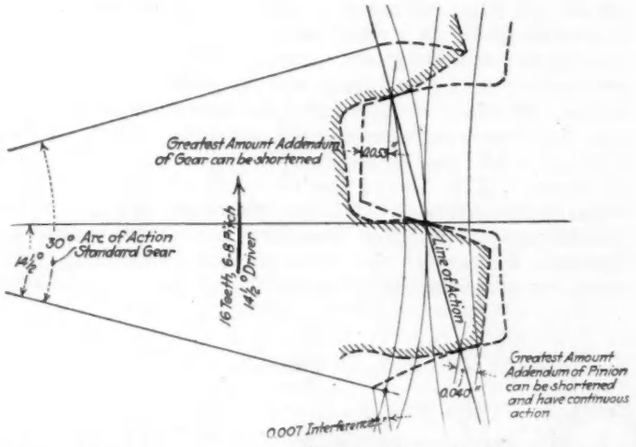

Greatest Amount Addendum of Gear can be shortened
0.005"
30° Arc of Action Standard Gear
14½°
16 Teeth, 6-8 Pitch 14½° Driver
Line of Action
Greatest Amount Addendum of Pinion can be shortened and have continuous action
0.040"
0.007 Interference

Fig. 10—Preliminary Layout That Is Made To Determine the Length of Tooth Necessary To Give a Complete Arc of Contact

generates, roughly, a flat in the tooth-form. If any of the teeth in the hob is high, a wider flat will be produced as shown in Fig. 9. Should the tooth-heights be correct and the hob be mounted in the machine with a run-out, a leaning tooth can be produced, depending on the sidewise setting of the hob with the gear. Also, should the hob be correct and the end-thrust collar in the hob spindle be out of parallel, giving the hob a slightly reciprocating motion with each revolution, an error can be produced that may compensate for the hob run-out or may add to it. Should the thrust collar at the rear end of the spindle be adjusted loosely so that the spindle may have end-play, the hob, as it cuts on one side, will be forced over and then back with each tooth of the gear and produce corresponding errors in the tooth-form. Again, if the gears in the hob-grinding spindle have inherent index-errors in them, or should the gears driving this gear be concentric or improperly spaced, their errors will be transferred to the different teeth of the gear being cut. Another important element in connection with the hob

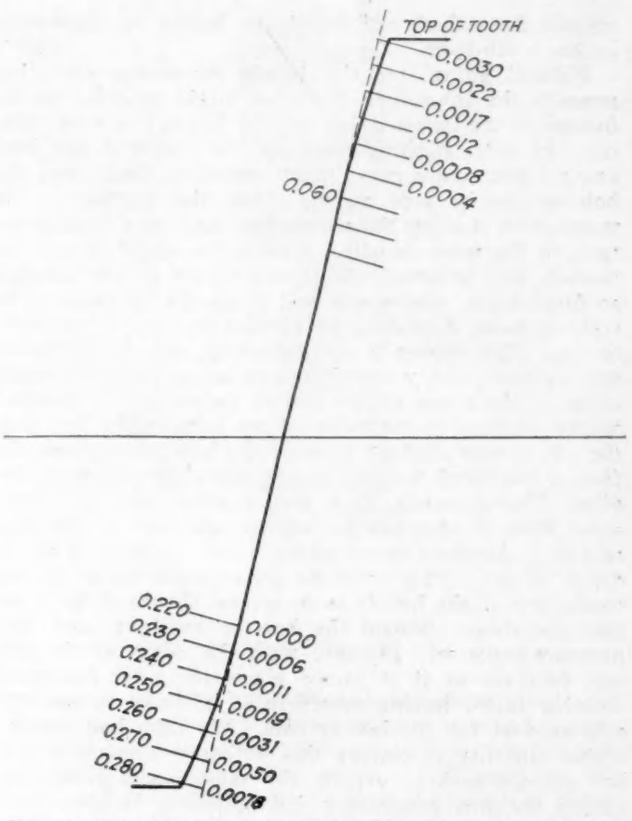


Fig. 11—Outline of the Hob Form Selected from the Layout Shown in Fig. 10

is the fit of the hob spindle. We will all agree that, should the hob spindle be tight for a certain portion of the revolution and loose for a certain other portion of the revolution, a sagging will occur in the driving-gear train which will be very detrimental to the tooth-form. Some hobbing machines are built so that the bevel gears in the hob drive-spindle give a thrust in the opposite direction to that given by a spiral pinion driving a hob-spindle gear. This permits a back-and-forth movement of a hob-

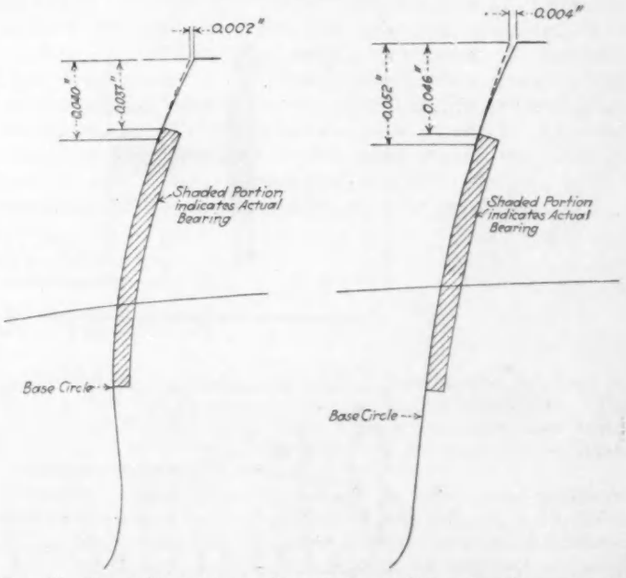


Fig. 12—Tooth-Forms of a Pair of Gears That Are Intended To Be in Contact with Each Other as Obtained from the Hob Outline Shown in Fig. 11

spindle drive-shaft and sometimes leaves its impression on the tooth-form.

Without going into the details concerning the other gears in the hob-spindle train, we might consider the influence of the thrust collar and the fits of the work spindle. In most hobbing-machines the bearings are kept fairly tight, and a great many operators insist that the hob spindle be kept warm. This also applies to the worm-shaft driving the wormwheel and, to a certain extent, to the work spindle. Unless the machines are extremely well adjusted, the thrust collars on the spindles so fitted score very easily and cause the spindles to be tight or loose, depending on various portions of its revolution. This causes a corresponding sag in the gears driving the index wormwheel and seems to be the main cause of the index errors already referred to. Another source of error is looseness in the gibs of the hob saddle. It is very difficult to move the hob slide across the face of the tooth without having some play between the gibs. This amounts, in a very similar manner, to the error that is obtained by having end-play in the hob spindle. Another contributing error to tooth-form is depth of cut. There will be considerable error in the tooth-form if the hob is sunk several thousandths of an inch too deep. Should the hob be straight-sided, the pressure-angle will increase with the depth of the cut and decrease as it is raised above the pitch-diameter. Another factor having considerable influence is the outside support for the hob spindle. We have had considerable difficulty in placing this outboard support of the hob spindle back in exactly the same place, giving us exactly the same condition of hob spindle as before.

When the work spindle is caused to rotate ahead of or behind its proper position, we necessarily have certain tooth-form errors in addition to index errors, and also in addition to those produced by the hob, its spindle and driving mechanism. There are conditions under which some of the errors referred to are counterbalanced by other errors. However, there are also conditions in which these errors accumulate. Considering the number of gears in a hobbing machine and the number of possibilities for errors outside of these gears, it is largely a matter of chance whether suitable combinations can be obtained to produce proper tooth-forms.

Tooth-Forms

Having all these liabilities to error in mind, the question often arises as to which is the noisiest tooth-form. Of the various kinds of gear that we have been able to cut and that we have had cut for us by a large number of different manufacturers, we have found that the errors from a general definite shape have apparently more to do with noise than any existing type of gear. The nearer to being correct a 14½-deg. pressure-angle-tooth gear is, the quieter it will be. We have never yet been able to secure 14½ and 20-deg. pressure-angle gears that had similar errors. The indications are, however, that the differences in sound due to the differences in the pressure-angle are very slight when compared with the differences in sound due to different errors in the same gear. We are using 14½ and 20-deg. pressure-angle gears regularly in production, and there seems to be little to choose between them.

The details of tooth-form are of some importance. Continuing with transmission gears, Fig. 10 shows the first layout which we make. This is with a view to determining the amount of tooth length necessary to give a 100-per cent arc of contact. With this information in hand, we select a hob form such as is shown in Fig. 11 and, using this on paper, we roll out two tooth-forms as shown in Fig. 12, one for each of the gears that are intended to be in contact with each other. The next step is to roll these gears on each other to determine the interferences, if any, and the amount that they are topped-off; then, if necessary, the hob form is modified and the same procedure carried through. When the hob form is established on paper in this manner, it is charted as shown in Fig. 13 and the hob supplier is asked to conform to this shape. Definite tolerances are given for the amount of variation from this form. On receipt of the hob, we inspect this form on a hob-checking apparatus very similar to that which we use for checking

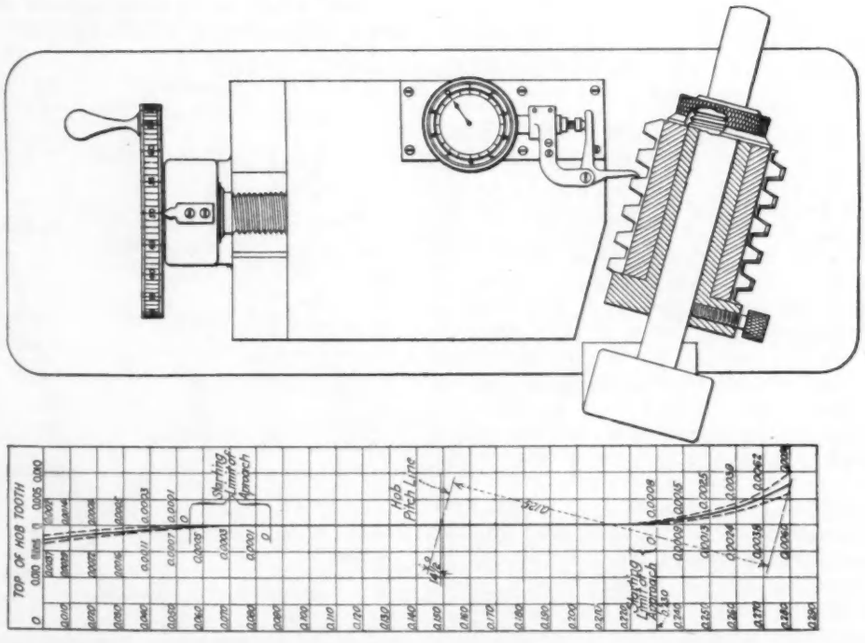


Fig. 13—Apparatus Used for Checking Hobs after Their Receipt and a Typical Diagram That Is Furnished Suppliers of Hobs Giving the Outline of the Hob Form

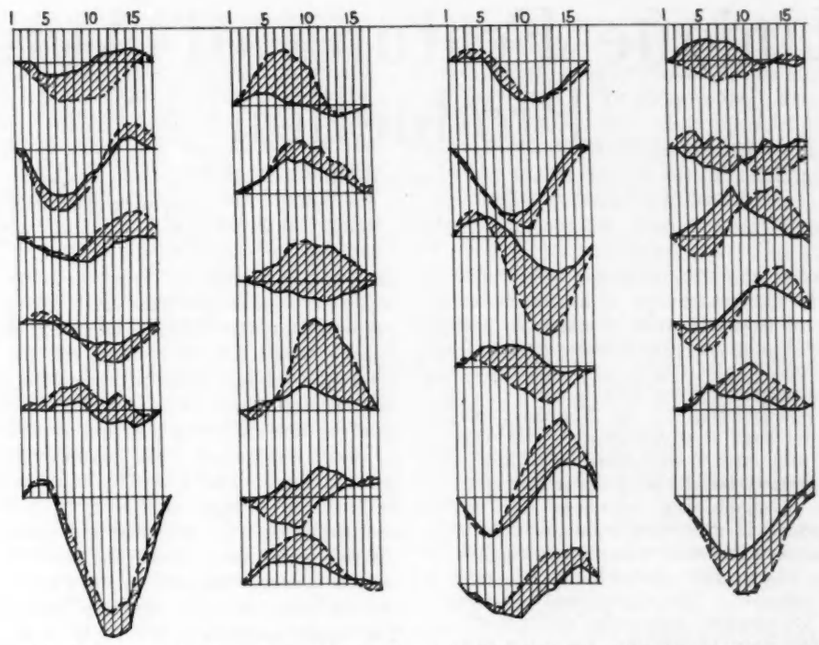


FIG. 14—CHART SHOWING THE VARIATION IN 24 GEARS, BEFORE AND AFTER HARDENING

gear-tooth forms, a drawing of which is shown in Fig. 13. If these conform to our standard requirement, it is expected that the hob will be satisfactory. We do not require any test of the hob in a cutting machine, because of the large number of errors that will be introduced by the machine, either in correcting hob errors or resulting in having an apparently correct hob rejected without cause.

HARDENING OF GEARS

Relative to the errors produced by hardening, we have prepared a number of charts showing the condition of the gear in the green and the condition of the gear in the hard. Fig. 14 shows 24 such gears and the variations occurring in them. Similar variations occur in gears mating with these gears. It will be seen that the cutting errors are considerable and that, at times, the hardening errors compensate for the cutting errors, while at other times the hardening errors go in the same accumulative direction.

Prof. John J. Keller gave a very interesting paper before the American Society for Steel Treating recently on why steel warps, showing what happens in a piece of steel when it is cooled and quenched. My impression from this paper is that freedom from warpage in steel is a matter of uniform hardening and cooling; also of removing all forging strains before machining. The question as to whether oil-treated steel is better than carburized is still unanswered as regards warpage. We have hardened more than 5000 gears of different brands of steel and carefully checked them. We find that there is very little difference in the warpage under the same hardening conditions.

GRINDING GEAR-TOOTH FORMS

The necessity for grinding gear-tooth forms depends largely on our ability to cut and harden gears, maintaining definite shapes. However, there is a large difference in the number of rejections that we have from gears ground by different processes. Our reports at this time show that out of 5000 gears ground by one method we have had a 14-per cent rejection. This is slightly greater than that which we have had from the hob gear without grinding. By another method of grinding of a similar number of gears, we have had less than a 0.5-per cent rejection, as well as more satisfactory gears. In the first case, four gears of the transmission were ground, and in the second case only two gears of the transmission were ground. All transmissions were passed by the same inspector and inspected to the same standards. We, of course, are looking forward with great interest to the continuation of our experiment on gear-grinding.

FAN DYNAMOMETERS

ON p. 45 of Vol. II. of the S. A. E. HANDBOOK is published information on the fan dynamometers. As Vol. II has not been issued since October, 1920, members of the Society joining since that date have not received copies of this volume containing general engineering information. Members using the volumes issued prior to 1920, however, should bear in mind that the note on p. 45a, reading "Radial setting figures appearing on the curves represent distance from center of hub to center of area of blades," is incorrect as the numbers on the curves actually indicate inches from the center of one blade to the center of the other. In using this information, it should be appreciated that results obtained by the use of fan dynamometers are highly unreliable. The Bureau of Standards has made tests which show that variations of as much as 20 per cent are possible under certain weather conditions.

Members desiring to supplement the information published in the Handbook on this subject should refer to a report on "Variable Torque Fan Dynamometers" issued by the National Advisory Committee for Aeronautics, an abstract of which appeared in *Automotive Industries* for March 24, 1921, and to p. 1688 of *Marks Mechanical Engineers Handbook.*

Some Unique Features of Automobile Production

By William Dunk[1]

Detroit Production Meeting Paper

THE author discusses the Franklin automobile designs in regard to the production of an air-cooled cylinder, the making of case-hardened crankshafts and the machining of duralumin connecting-rods, commenting also upon experiments made with a view to producing hot-swaged rear-axle drive-shafts.

The unit-cylinder construction of the Franklin engine, with the cylinder-head cast integrally with the body of the casting and steel cooling-fins cast in the outer surface of the cylinder-wall, presents a different problem from that of machining cylinders cast-in-block, and the procedure is described in some detail, inclusive of the machine tools used. The case-hardened crankshaft and the duralumin connecting-rod are treated in a similar manner. The experiments in the manufacture of a hot-swaged rear-axle drive-shaft have extended over the past 18 months, but have not yet justified quantity production by this method.

ALL engineers realize that there are a large number of problems to be met in the production of automobiles, and that the majority of these problems are much the same with the H. H. Franklin Mfg. Co. as with any other. I will endeavor, therefore, to confine this discussion to the following topics that are somewhat unique in Franklin automobile designs; namely, (*a*) the production of an air-cooled cylinder, (*b*) the making of case-hardened crankshafts, (*c*) the machining of a duralumin connecting-rod and (*d*) experiments with a hot-swaged rear-axle drive-shaft.

Production of the Air-Cooled Cylinder

The design of the Franklin engine necessitates the use of unit-cylinder construction rather than the regular block-cylinder design. This fact makes the production of the cylinder a somewhat different problem from that of machining a block casting. In the Franklin cylinder, the head is cast integrally with the body of the casting and steel cooling fins are cast in the outer surface of the cylinder-wall.

The foundry practice in connection with the production of these castings is of interest. The mold is composed of three distinct sections, the lower section being an assembly of dry-sand cores for making the head of the cylinder and the valve ports. Next above this section is a green-sand section containing the cooling fins, and the upper section of the mold is a green-sand section for the neck of the cylinder between the cooling fins and the bottom flange, the cylinder of course being cast in an inverted position. The core assembly for the head of the cylinder is made up of five unit-cores that are made separately and assembled by pasting-in an accurate pasting jig so that the valve-port cores are held accurately in position while drying. The valve-port cores are provided with prints that are used to locate the upper end of the main cylinder-barrel core; for this reason it is necessary to be very accurate in the location of the port cores.

In making the green-sand section containing the cooling fins, the cylinder pattern is milled longitudinally with the proper number of grooves to receive the vertical cooling-fins. These grooves are 1/16 in. in depth, which corresponds to the amount that the fin is imbedded in the cylinder-wall. This section of the mold is made on an air-operated jar ramming-machine provided with a sand hopper directly above the mold. A very small amount of sand is used in this section of the mold, not over 2 in. of sand being provided outside of the diameter of the cooling fins. The fins are held in place in the cylinder pattern by simply wrapping a piece of soft iron wire around them and twisting the ends tight. This wire remains in the mold until the casting has been made, and serves to keep the section of green sand that lies within the cooling fins from sagging below the rest of the mold. The upper section of the mold is also made in green sand and is located on the center section by taper dowels. The last operation in making the mold is the lacing of the cylinder-barrel core and the strainer-gate core through which the metal is poured. The cooling fins are made of ordinary cold-rolled strip-steel which has been tinned and folded to a right angle about 5/16 in. from one edge. These cooling fins, when placed radially about the cylinder pattern, form a completely enclosed air-jacket.

The greatest source of casting loss that we experience in making this piece is caused by loose flanges. Unless the flanges extend at least 1/16 in. below the surface of the cast iron, they will not be properly held in place.

The analysis of iron used in making the Franklin cylinder is approximately as stated in Table 1.

TABLE 1—ANALYSIS OF IRON USED IN THE FRANKLIN CYLINDER

Constituents	Per Cent
Silicon,	2.00
Phosphorus,	0.50
Manganese,	0.50
Sulphur,	0.10
Carbon,	3.25

The machining operations are as follows: The cylinder casting is first mounted in a chucking fixture on the table of a small vertical boring-mill, and the bottom flange is faced and counterbored. This counterbore is used for locating the cylinder while drilling the four hold-down holes on the second operation. These hold-down holes are used for locating the cylinder for the boring operation, which is performed next. Our present method of boring the cylinder consists of making four separate cuts into the cylinder under a single-spindle Baker boring-machine. The necessity of making this number of cuts is caused by the fact that the cylinder-wall has no support to prevent its being split, if a heavy cut were attempted. The thickness of the metal of the finished cylinder between the cooling fins and the cylinder bore is only 3/16 in.

A new cylinder-boring machine has just been completed and tested which makes five separate cuts in the cylinder, operating on five separate cylinders simul-

[1] Production manager, H. H. Franklin Mfg. Co., Syracuse, N. Y.

taneously. The first spindle of the machine rough-bores, and removes about ⅛ in. of metal from the cylinder-wall. The second spindle rough-bottoms the cylinder to within 1/32 in. of the finished depth. The third spindle semi-finish-bores, and leaves 0.035 in. of metal for finish-boring. The fourth spindle finishes the bottom of the cylinder to depth, and the fifth finish-bores to grinding size. We are leaving about 0.007 to 0.010 in. of metal for grinding at present, and we may find it possible to reduce this to a maximum of 0.005 in. The fact that we are dealing with unit cylinders allows us to center the cylinder very accurately for the grinding operation. The complete boring operation as performed by the new cylinder-boring machine occupies about 1½ min. per cylinder.

The cylinder is ground on the standard type of Heald cylinder-grinding machine equipped with a special small table for unit-cylinder work. It is held against an ordinary angle-plate grinding-fixture, and is located with a plug through a master bushing inserted in the grinding fixture. Cooling water is flowed on the outside of the cylinder during the grinding operation. The valve ports in the cylinder are reamed holes about 2½ in. in diameter and 1 in. in depth. They are held to a limit of 0.001 in. to provide a sufficiently tight joint with the extension pieces that attach them to the suction and exhaust manifolds. A special double-head machine was built to bore and ream the port holes as accurately as this.

Briefly, this machine consists of two three-spindle movable-heads operating toward a central holding-fixture on which four cylinders are mounted; there are three working positions and one loading position. The central fixture is arranged so that it can be indexed at the end of each stroke of the machine. The first pair of working spindles rough-bores the port holes to within 0.035 in. of the finished size. The second spindle does not remove any metal from the diameter of the hole but simply performs the bottoming operation. The third pair of spindles finish-reams the holes to size. By arranging the machine to operate on both ports at once, the thrust of the cutters has been neutralized almost exactly, because the ports lie nearly opposite each other across the cylinder casting. The cutters used in this machine for all operations are a standard type of shell-end mill, and they seem to give very satisfactory results. It is not necessary to change cutters oftener than 1 set to each 350 cylinders. The capacity of this machine is one complete cylinder every 40 sec. A particularly rapid means of loading and unloading the cylinder is necessary, to allow the operator to perform these operations during the cycle of the machine.

The valve-stem guides are separate castings machined complete and pressed into place in the cylinder casting, the final reaming operation being performed in the valve-stem holes after the guides are pressed-in. The seating and other minor operations on the cylinder are performed in the usual manner and do not require any special mention.

The Case-Hardened Crankshaft

Another interesting manufacturing detail is the Franklin case-hardened crankshaft, which we have been producing for the past 2 years. The primary object in incorporating this type of shaft is naturally to prolong the life, so far as possible, of the bearing surfaces. Preliminary experiments developed the fact that, if a crankshaft can be hardened to a scleroscope hardness of between 80 and 100, the life of both the main bearings and the crankpin bearings can be prolonged considerably. In some of our test cars as much as 50,000 miles was covered without any adjustment of the bearings being necessary.

In putting a proposition of this kind into production, a number of problems had to be met concerning which very few data were obtainable. The crankshaft forging has a carbon-content of from 0.15 to 0.25 per cent and the required depth of carburization is specified at 1/16 in.

In the manufacture of this shaft, the forging is first machined to within 0.040 or 0.045 in. of its finished diameter, with the exception of the flange and threaded ends which are left in the rough. The forging is then ready for the copper-plating operation, which is applied to prevent carburization of all parts of the shaft except the bearing surfaces. Several ideas were tried out in connection with preventing the copper-plating of these surfaces during the process, and we finally adopted the method of wrapping all bearings with pure gum rubber held in place with clips. This affords a very simple means for preventing the copper-plating of the surfaces; the rubber bands are easily removed and can be used again and again.

In carburizing the shafts they are packed, three in a box, in nickel-alloy carburizing-boxes, using either Q alloy or thermalloy. The approximate size of the carburizing boxes required is 12 x 44 in., and about 18 in. deep. A liberal amount of carburizing material is provided around each shaft. The boxes are sealed with the regulation cement and are loaded into the carburizing furnaces by a specially constructed charging truck, each furnace-chamber having a capacity of six of these boxes and each box having a total weight, loaded, of about 800 lb. The carburizing period required to get the required depth of penetration is between 22 and 24 hr. The use of nickel-alloy boxes seems to have a decided tendency to reduce the number of hours required to get this penetration.

At the end of the carburizing process, the shaft is returned to the crankshaft-machining department; the flange end and the threaded end are machined, roughly, to the finished dimensions, this operation having the result of removing all of the case-hardened surface produced by carburization. This allows for the machining of these parts after the shaft has been hardened.

The shaft is then returned to the heat-treating department and reheated in a gas-fired furnace to a temperature of 1420 deg. fahr. To keep the shaft as nearly straight as possible during the quenching operation, a hinged type of quenching die was devised that permits rapid loading for the purpose of quenching the shaft as soon as possible after removing it from the furnace. This disc is of very heavy construction, and is handled by an air hoist. The shaft is quenched in water at main temperature, and comes from the quenching operation usually not over 1/16 in. out of straight. It is necessary to straighten the shaft carefully at this point, before proceeding with the semi-finish and finish-grinding operations.

We have found it necessary to divide the finish-grinding operations into semi- and finish-grinding so as to use a rather coarse soft grinding-wheel for the semi-finish grinding in bringing the shaft to the finished size. The last 0.005 in. of metal is removed from the shaft with a very fine soft wheel to get the required smoothness of finish.

One somewhat uncertain feature that we always meet in the production of this shaft is the amount of shrinkage that will take place in the hardening operation. It is necessary to anticipate this, and to provide for it in machining the soft shaft. Our experience has shown that this shrinkage varies to a considerable degree. Our normal allowance in the overall length of the shaft and

the spacing of the bearings is about 0.035 in. for a 5½-in. center-distance between cylinders. We find that this allows us to finish the shaft within the limits required for length, in the majority of cases.

Our scleroscope-hardness requirements of 80 to 100 give us a shaft that has exceptionally long life. I neglected to state, however, that the point of contact of the quenching die with the shaft produces very small soft spots; but they do not seem to interfere with the general durability of the bearing surfaces in any way.

The Connecting-Rod

The Franklin connecting-rod is an aluminum-alloy forging, either of duralumin as furnished by the Baush Machine Tool Co. or of 17-S alloy furnished by the Aluminum Co. of America. This aluminum alloy contains the elements given in Table 2.

TABLE 2—ANALYSIS OF ALUMINUM ALLOY IN THE FRANKLIN CONNECTING-ROD

Constituents	Per Cent
Copper,	3.50
Manganese,	0.20
Magnesium,	0.25
Iron, not more than,	0.75
Silicon, not more than,	0.75

The elastic-limit of the forging is between 30,000 and 35,000 lb. per sq. in.; the ultimate-strength is between 50,000 and 55,000 lb. per sq. in.; and the elongation is from 15 to 25 per cent.

The forging is heat-treated by heating it to 920 or 940 deg. fahr., and is quenched in boiling water. It is then allowed to age about 1 week to bring it to a scelerscope hardness of 90 to 100.

We use about the following procedure in machining the duralumin forged connecting-rod. The forging is first coined under a heavy toggle-press, bringing the length of the wristpin and crankpin ends to within 0.025 in. of the finished size. It is then loaded onto a special fixture on a Blanchard grinding machine that has alternate roughing and finishing stations. It is ground first on one side, then removed from the roughing station, moved to the next adjacent station and finish-ground on the other side. A special multiple-spindle boring-and-reaming machine having an indexing table is used for boring and reaming the wristpin and crankpin holes. The operations which this machine performs are to rough-drill, rough-ream and semi-finish ream.

The action of the forging in these machining operations is similar to that of mild steel. The power required to drive the drills and reamers is approximately the same as the power we employed formerly with our steel connecting-rod, and the cutting speed does not exceed that for steel by more than 25 per cent. In machining this forging, it surprised us to find that it machined so nearly like a steel rod. We had anticipated being able to machine this rod at a much greater speed than we found possible. The extreme toughness and wire-like quality of the chips are convincing proof that the metal with which we are dealing is something more than ordinary aluminum.

The forging is next drilled for the rod bolts and is then put over a multiple fixture for parting the cap from the rod and facing the seats for the bolts and nuts. A broaching operation follows for providing slots in which the lower-end babbitt blocks are locked. These babbitt blocks are forced into place and broached to their semi-finished size in an ordinary two-spindle broaching-machine. The finishing of the wristpin end of the rod has been rather difficult, because it is impossible to ream a sufficiently good wristpin hole in the bearing end. Our present method of handling this operation is to semi-finish ream this hole, removing about 0.025 to 0.030 in. of the diameter, then to use a final reaming operation that removes about 0.005 in. and a final burnishing operation that is done with a high-speed burnishing-tool under a copious flow of regular medium automobile motor-oil. We find that by leaving a uniform amount of about 0.0005 in. on the diameter we can produce a highly burnished surface in the wristpin hole and maintain very uniform size. However, to obtain the fits required by our engineering department between the wristpin and the connecting-rod, it is necessary to do a certain amount of wristpin selection. When it is finally finished, the alignment of this rod is held to a limit of ±0.005 in., measured between the ends of arbors 12 in. long.

The finishing to size of the lower end of the rod is left until it is ready for assembly with the crankshaft. This operation is performed on a special type of boring machine in which we use boring bars of five separate diameters, increasing their diameters in steps of 0.00025 in. from the minimum size required up to the maximum; this allows us to cover the entire range in the crankpin bearings with the proper amount of oil clearance. At this time we use an oil clearance of about 0.001 in. for this bearing.

The weight of the finished connecting-rod runs very uniform, it being necessary to maintain but two standards of weight to keep within the limits specified by our engineers, which are a variation in weight of any two connecting-rods on an engine of not more than 0.2 oz.

Hot-Swaged Rear-Axle Drive-Shaft

We have been experimenting for the past 18 months with the production of drive-shafts by the hot-swaging process but, up to the present, we have not felt justified in following this method of manufacture in large production quantities. The saving in the amount of material used to make a rear-axle drive-shaft, and the reduction in the amount of labor required by this method, make it a very attractive proposition, and we are not inclined to give up the idea without a struggle.

Troubles have developed in the source of our experiments. We have been unable to obtain a swaging machine of a construction heavy enough to withstand the load put upon it by an operation of this kind. We had experienced continued breakdowns of our original machined, and we have practically rebuilt it. The problem of producing a swaged shaft, without having the finished product show any twisting effect whatever from the swaging, has been difficult.

In performing the second swaging operation, which consists of necking the shaft below the diameter of the splined end, we have not been able to reheat for this operation without excessive scaling, which mars the appearance and reduces the strength of the finished shaft. The procedure that we are following at this time requires three separate swaging operations and three separate heats for making them. If it were possible to take the shaft directly from the first swaging operation and bring it up to swaging temperature again, which is about 2000 deg. fahr. in our case, and to perform the second or necking operation immediately, we believe we could reduce the amount of twisting and the amount of scaling that we experience at present. The third swaging operation, that of producing the taper and the threaded end, is comparatively simple and gives us no trouble.

We are now considering the installation of a much heavier swaging machine. We find it can be built to order for us. We are considering also the installation of a second furnace to be used as a booster between swaging operations Nos. 1 and 2. With this proposed new equipment, we should be able to produce a very satisfactory product. We would be very glad, however, to learn of the experiences of some of the other automobile companies in producing a hot-swaged rear-axle driveshaft.

EDUCATION FOR HIGHWAY ENGINEERING AND TRANSPORT

THE second national conference on education for highway engineering and highway transport was held in the City of Washington, on Oct. 26 to 28, under the auspices of the Highway Education Board. This Board has headquarters at the City of Washington and is composed of representatives of those agencies of the Government and industry having an intimate contact with and interest in the construction and maintenance of American highways. The personnel of the Board is as follows: John J. Tigert, United States Commissioner of Education, chairman; Thomas H. MacDonald, chief of the bureau of public roads, Department of Agriculture; Roy D. Chapin, vice-president of the National Automobile Chamber of Commerce; H. W. Alden, representing the Society of Automotive Engineers; Harvey S. Firestone, representing the Rubber Association of America; Col. F. C. Boggs, Corps of Engineers, United States Army, War Department; F. L. Bishop, dean of the School of Engineering, University of Pittsburgh, secretary of the Society for the Promotion of Engineering Education; and a representative to be selected by the American Association of State Highway Officials.

An important object of the conference was to discuss means of meeting the public demand for trained men in the construction and operation of highways, and for this reason the work of the 11 educational committees must be considered as constituting a vital part of the meeting. The program allotted much time to the sessions of these committees, whose final recommendations on details of curricula may be expected to be ready for announcement in about 4 months. One afternoon session of the general meeting was devoted to the subject of the content of prospective courses on highway engineering and highway transport in the engineering schools.

On the general program there were at least two addresses of direct appeal to the automotive engineer, and, curiously enough, both of them emphasized the success of the British system of collection and distribution of goods by motor truck. The first address, by Frederick C. Horner, on European and British Road Transport Practice, described more particularly the types and design features of the British trucks, while the second, by W. H. Lyford, on Cooperation versus Competition Between Motor Trucks and Railroad Transportation, discussed the application of the British system to American conditions. Mr. Lyford, who is vice-president of the Chicago & Eastern Illinois Railroad, is of the opinion that American method applied to the British system can make the motor-trucking company a valuable ally of the railroad with profit and advantage to both parties. This view was presented as the conclusion of a remarkably able and profound study of the situation.

The other addresses of the general meeting dealt with economic and sociological features of highway transport and with various phases of research. It is interesting to note that the need for more investigation in the field of automotive engineering is recognized as a part of the problem of highway transport, though provision for the prosecution of automotive research is not contemplated by the existing organization for highway research. This is implied, for instance, in the address of Dr. W. K. Hatt, director of the advisory board on highway research of the National Research Council, and is otherwise expressed in Prof. Arthur Blanchard's proposal that truck producers create for this purpose an organization similar to the Portland Cement Manufacturers' Association or the American Zinc Institute.

Research was a topic that pervaded the entire program. This may find an explanation in the fact that a large proportion of the 225 delegates to the conference represented engineering schools. Not the least effective of the arguments for research were the two excursions to the Arlington Experimental Farm and to College Park, University of Maryland. The arrangements for these excursions were excellent, and practically every delegate was enabled to get a good insight into the progress of the investigations. The motion pictures of the Pittsburg, Cal., road-test experiments, as explained by the engineer in charge of the work, also created favorable comment in this connection, but probably the most impressive illustration of the value of research was afforded in the reading of the circular just issued by the Illinois State Highway Department, announcing new specifications for the cross-section of concrete pavements. This modified cross-section is a direct result of the Bates experimental road work, and its timely appearance on this occasion had a dramatic effect, because the new cross-section will provide a pavement that not only is better able to support trucks loaded to the legal limit, but will also reduce the cost per mile.

The National Automobile Chamber of Commerce must be credited with having added much to the interest of the general meeting, not only by providing conspicuously able speakers at strategic points of the program, but also by sponsoring the exhibit of the Eugene B. Clark collection of original paintings dedicated to the automobile industry, and entitled The Spirit of Transportation. This exhibit attracted the attention of the general public, as well as that of the delegates to the convention.

The Group-Bonus Wage-Incentive Plan

By E. Karl Wennerlund[1]

Detroit Production Meeting Paper *Illustrated with* Charts

THE author states that the purposes of every plan of wage incentive are to stimulate the worker to a greater effort than is generally obtained on a straight day's-work basis; to reward him somewhat in proportion to his effort; and the gaining of other advantages such as greater attention to conditions that curtail production, more uniform labor costs and the elimination of inefficient employes. He states further that nearly all industries engaged in repetitive work are now on an incentive basis.

After outlining the most successful wage-incentive plans and enumerating some of the conditions that must be met, inclusive of four specific fundamental principles of industry that are stated, the group-bonus plan is explained and the application of group standard-time is discussed at some length, supplemented by tabular data. Experience with grouping is then related and conditions favorable to grouping are mentioned. The summary states that the group plan, like any other wage-incentive plan, has for its main objective intensive production.

THE subject of wage incentive is an important one to factory executives. During the past few years it has received considerable study. Nearly all factories now have their own specialists who make time-studies of operations and recommend or set production rates. Even where outside industrial engineers are employed for this purpose their services are regarded as temporary, with the idea of developing a local staff as rapidly as possible to handle the regular routine. The question of the particular form of wage incentive, therefore, becomes important. No single plan can be recommended for all factories. Local conditions and the character of product will have to govern in each case. It is, of course, desirable to have an incentive plan handled as simply as possible with a minimum of clerical effort, and with the least liability to error in quantity check.

The purposes of every plan of wage incentive are the same:

(1) To stimulate the worker to a greater effort than is generally obtained on a straight day's-work basis

(2) To reward him somewhat in proportion to his effort

(3) The gaining of other advantages such as the attention given to conditions that curtail production, more uniform labor costs and the elimination of inefficient employees

The result has been that nearly all industries engaged in repetitive work, with very few exceptions, are on an incentive basis. Until recently, the plan followed has been generally of the type known as the individual-effort plan, with either straight piece-rates or some form of premium or bonus based on time measurements. Grouping of employes was not often resorted to, except for such operations as made it impracticable to keep track of the output from individuals. However, the principles of grouping, with a division of the earnings among several workers, have been well known and have been applied for a long time to such work as steam hammering, riveting and handling materials; but, for work done distinctly by an individual, the usual practice has been to set a rate for that particular detail operation and to pay the individual worker for his production at that operation. The application of grouping of employes throughout an entire factory organization, wherever work is measurable, is rather a new development in industry.

In automobile factories and most accessory plants, the character of product is such as to make parts move progressively from one operation to the next to a very great extent. Processes have been standardized and divided into minute detail and, being of such repetitive nature, operations have been segregated into progressive flow-lines wherever practicable. This was a natural result from the desire to minimize the expense and delays incident to transporting materials in process. The consequent physical arrangement of these factories has had an important bearing on wage-incentive problems.

It must be taken into account that an individual-effort system of payment to workers entails considerable routine expense. The earliest method, after rates had been set, was for each worker to make out a report at the end of the day stating how many pieces he had completed under each operation; or a shop checker came for this information and inserted the quantities on a prepared form. A systematic check on quantities after each operation was, of course, impracticable with large-volume production unless some special means were provided; so, there was much opportunity for collusion between shop checkers and workers and there is no doubt that many took advantage, particularly on parallel operations, of the management's inability to obtain an actual check. One safeguard often resorted to was to discourage the worker from turning in more than a certain maximum rate of earnings; but, even then, it was no guarantee that amounts up to the maximum might not be fraudulently claimed. It was just about as logical to continue to operate a factory under these conditions as it would be to eliminate the time-clocks at the entrance gate or to do away with cash-registers. To do so would require a lot of faith in human nature. There is no doubt that the time-clock and cash-register have both helped many men to remain honest.

Well-organized factories, operating under the individual-effort plan in recent years, have developed various methods for obtaining physical checks on quantities after each operation. One of these is to have a set of coupons, numbered serially, attached to each large piece or to a container of a pre-determined number of pieces. The worker is then required to present these coupons as evidence of quantities. By another method known as the lot system, the worker obtains a job ticket for each lot. The start and finish time of the operation for each lot is stamped on the job ticket which bears the same serial number as the lot quantity. Later, when these tickets come to the office, each is checked off on a master lot-card, then extended and credited to the individual account.

These methods are all good in theory. No doubt they pay their way when compared with having no definite check, and they may be entirely practicable in small shops or for certain classes of work, but let us consider the

modern automobile factory. The lot method not only ties up a large inventory of parts in process, but to check off and extend thousands of job tickets, daily, also involves a great expense. One such factory had in excess of 25,000 job tickets for each working day, each one of which had to be carefully audited as to quantity and rate, extended for total and then credited to the account of the employe.

It was the desire to simplify the factory routine, and to escape from this mass of detail management without sacrificing the principles of quantity checks, that prompted the development of the group-bonus plan. Certain fundamental principles of industry had to be recognized and taken into account. The most important of these are that

(1) Under any incentive plan, the worker must continue to maintain an individual interest in the rate of production

(2) He should be guaranteed a satisfactory hourly wage

(3) He should be paid for all time saved instead of for only a part of it

(4) He should be able to compute his own earnings, accurately and quickly

Under the group plan, it would not make much difference in theory whether a fixed group-price were established or some form of bonus or premium payment were used. Our past experience, however, indicated that it was preferable to set standards of output in *time* rather than in money value. With changing labor scales, we should then have no disturbance of the existing time-standards, but would make the adjustments on the hourly-base wage-rates on which bonus earnings are computed. Time standards also have the advantage of a very flexible wage-rate, by which means new employes can enter our service at a minimum rate subject to advancement through length of service and personal ability. Such revision in wage rates can be made at any time without in the least disturbing shop standards or affecting other employes.

TABLE 1—BONUS TABLE FOR PRODUCTIVE WORKERS

Percentage Efficiency	Bonus	Percentage Efficiency	Bonus	Percentage Efficiency	Bonus
75	1.0	117	40.4	159	90.8
76	1.6	118	41.6	160	92.0
77	2.2	119	42.8	161	93.2
78	2.8	120	44.0	162	94.4
79	3.4	121	45.2	163	95.6
80	4.0	122	46.4	164	96.8
81	4.6	123	47.6	165	98.0
82	5.2	124	48.8	166	99.2
83	5.8	125	50.0	167	100.4
84	6.4	126	51.2	168	101.6
85	7.0	127	52.4	169	102.8
86	7.6	128	53.6	170	104.0
87	8.2	129	54.8	171	105.2
88	8.8	130	56.0	172	106.4
89	9.4	131	57.2	173	107.6
90	10.0	132	58.4	174	108.8
91	11.0	133	59.6	175	110.0
92	12.0	134	60.8	176	111.2
93	13.0	135	62.0	177	112.4
94	14.0	136	63.2	178	113.6
95	15.0	137	64.4	179	114.8
96	16.0	138	65.6	180	116.0
97	17.0	139	66.8	181	117.2
98	18.0	140	68.0	182	118.4
99	19.0	141	69.2	183	119.6
100	20.0	142	70.4	184	120.8
101	21.2	143	71.6	185	122.0
102	22.4	144	72.8	186	123.2
103	23.6	145	74.0	187	124.4
104	24.8	146	75.2	188	125.6
105	26.0	147	76.4	189	126.8
106	27.2	148	77.6	190	128.0
107	28.4	149	78.8	191	129.2
108	29.6	150	80.0	192	130.4
109	30.8	151	81.2	193	131.6
110	32.0	152	82.4	194	132.8
111	33.2	153	83.6	195	134.0
112	34.4	154	84.8	196	135.2
113	35.6	155	86.0	197	136.4
114	36.8	156	87.2	198	137.6
115	38.0	157	88.4	199	138.8
116	39.2	158	89.6	200	140.0

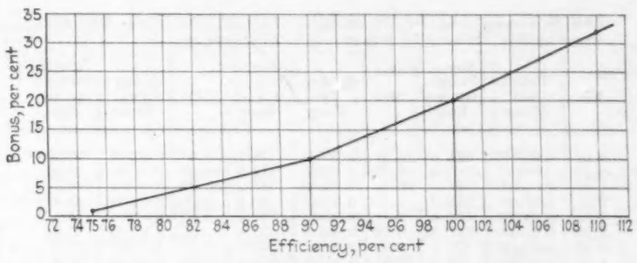


FIG. 1—CHART SHOWING THE RELATION BETWEEN EFFICIENCY AND THE PERCENTAGE OF BONUS

On the other hand, a fixed group-price in money value would offer some very definite objections if adopted as a general plan, since the

(1) Division of the earnings among the group members would have to be on an even basis per hour worked; or

(2) Group earnings would have to be pro-rated on the basis of the hours worked by the individuals, multiplied by the predetermined base-rates. A change in an individual base-rate would affect all other members of the group, or the transfer of an employee with a different base-rate would upset their relative earnings

In regard to a uniform base-rate for all members of a group and their division of the earnings purely on an hourly basis, it is obvious that this method would prove unsatisfactory as a general policy; because, even in the same group, some operations may require more skill than others. Then there is the added consideration that employes who have been long in the service should be on a higher wage plane than beginners. The standard-time bonus-table as adopted offers every facility for differentiating between base-rates and permits a flexible system to suit the varying labor and factory conditions.

Table 1 is based on the theory that a bonus incentive of 20 per cent should be paid to a worker, in addition to his hourly base-rate, for attaining an efficiency of production of 100 per cent; and that, for a greater speed of production, he should be paid for all time saved. The table is constructed so as to give a gradually disappearing percentage of bonus, as efficiency of production falls below the standard. During the past 11 years, this bonus table has been used extensively in our plants; first as an individual-effort incentive and, in later years, as a group incentive. A graphic chart showing the relation between efficiency and percentage of bonus is reproduced in Fig. 1.

The standard time of a single operation is determined by time-study. For this purpose the decimal stop-watch is used. By standard time is meant the time required by the average competent workman, taken repeatedly, over an extended period, eliminating lost and waste time. It will contain allowances for legitimate tool-changes and for rest and delay to the worker. It is our practice, for convenience, to designate standard time in decimal hours rather than in minutes. For this purpose a conversion table has been devised and is presented as Table 2, because the detail time-study is observed in decimal min-

TABLE 2—EQUIVALENTS OF MINUTES IN DECIMAL HOURS

Min.	Equivalent Hours	Rate of Production per Hour	Min.	Equivalent Hours	Rate of Production per Hour	Min.	Equivalent Hours	Rate of Production per Hour
0.01	0.000167	6000.0	0.34	0.00567	176.5	0.68	0.01133	88.2
0.02	0.000333	3000.0	0.35	0.00583	171.4	0.69	0.01150	87.0
0.03	0.000500	2000.0	0.36	0.00600	166.7	0.70	0.01167	85.7
0.04	0.000667	1500.0	0.37	0.00617	162.2	0.71	0.01183	84.5
0.05	0.000833	1200.0	0.38	0.00633	157.9	0.72	0.01200	83.3
0.06	0.001000	1000.0	0.39	0.00650	153.8	0.73	0.01217	82.2
0.07	0.001167	857.1	0.40	0.00667	150.0	0.74	0.01233	81.1
0.08	0.001333	750.0	0.41	0.00683	146.3	0.75	0.01250	80.0
0.09	0.001500	666.7	0.42	0.00700	142.9	0.76	0.01267	78.9
0.10	0.001670	600.0	0.43	0.00717	139.5	0.77	0.01283	77.9
0.11	0.001830	545.5	0.44	0.00733	136.4	0.78	0.01300	76.9
0.12	0.002000	500.0	0.45	0.00750	133.3	0.79	0.01317	75.9
0.13	0.002170	461.5	0.46	0.00767	130.4	0.80	0.01333	75.0
0.14	0.002330	428.6	0.47	0.00783	127.7	0.81	0.01350	74.1
0.15	0.002500	400.0	0.48	0.00800	125.0	0.82	0.01367	73.2
0.16	0.002670	375.0	0.49	0.00817	122.5	0.83	0.01383	72.3
0.17	0.002830	352.9	0.50	0.00833	120.0	0.84	0.01400	71.4
0.18	0.003000	333.3	0.51	0.00850	117.6	0.85	0.01417	70.6
0.19	0.003170	315.8	0.52	0.00867	115.4	0.86	0.01433	69.8
0.20	0.003330	300.0	0.53	0.00883	113.2	0.87	0.01450	69.0
0.21	0.003500	285.7	0.54	0.00300	111.1	0.88	0.01467	68.2
0.22	0.003670	272.7	0.55	0.00917	109.1	0.89	0.01483	67.4
0.23	0.003830	260.9	0.56	0.00933	107.1	0.90	0.01500	66.7
0.24	0.004000	250.0	0.57	0.00950	105.3	0.91	0.01517	65.9
0.25	0.004170	240.0	0.58	0.00967	101.4	0.92	0.01533	65.2
0.26	0.004330	230.8	0.59	0.00983	101.7	0.93	0.01550	64.5
0.27	0.004500	222.2	0.60	0.01000	100.0	0.94	0.01567	63.8
0.28	0.004670	214.3	0.61	0.01017	98.4	0.95	0.01583	63.2
0.29	0.004830	206.9	0.62	0.01033	96.8	0.96	0.01600	62.5
0.30	0.005000	200.0	0.63	0.01050	95.2	0.97	0.01617	61.8
0.31	0.005170	193.5	0.64	0.01067	93.8	0.98	0.01633	61.2
0.32	0.005330	187.5	0.65	0.01083	92.3	0.99	0.01650	60.6
0.33	0.005500	181.8	0.66	0.01100	90.9	1.00	0.01667	60.0
			0.67	0.01117	89.6			

utes, while the standard time is converted into hours.

Group standard-time is obtained by totalling the individual standard-times for all operation embraced by a group. It thus always appears as one-man hours, independently of how many workers may subsequently be assigned to a designated group. All detail operation standard-times and the total group standards are listed on a shop routing for ready reference as shown in Fig. 2.

Application of Group Standard-Time

To obtain a clear idea of the application of group standard-time and its use as a basis for a wage-incentive plan, let us consider a single production line manufacturing and assembling pistons in an automobile-engine plant. This unit has its operations arranged in sequence, in what is known as a progressive line of manufacture. It starts with the rough-machining of the casting and ends with the finished product with piston-rings inserted. The production line includes both machine and bench-assembly operations. Parts move without a break from one operation to the next in a steady flow. Some operations may be done by a single workman; others may have several workers in parallel, depending on the volume of production and with what detail it is practicable to subdivide operations.

It is proposed to keep no check on quantities passing intermediate operations or on those completed by individual workers on operations in parallel but to give credit for finished pistons passing final inspection. This particular production line may be located in a department also producing connecting-rods or any other line of manufacture. Such a line embraces a production unit of the factory, and is technically known under the group plan as a *division*. Its workers are primarily interested in the production of pistons; they know little or nothing about conditions on the connecting-rod line and should not be grouped with it.

A study of the piston-line division indicates that there are two distinct classes of machine work, besides bench operations. The machine operations are lathe turning and grinding. There are thus three major operations, and the workers group themselves naturally into these three classes of work. It is thus seen that a division may consist of any number of distinct groups. They are bound together because they receive credit only for the finished product at the end of the division, while they are distinct and independent so far as efficiency measurement and bonus reward are concerned. A group is debited with the actual man-hours of the workers in the group and is credited with the group standard-hours multiplied by the number of pieces passing the division. That is, the quantity count is not taken at the group operations but at the end of the division of work. The ratio between standard hours and actual hours constitutes the efficiency measurement for which a bonus percentage is paid. For convenience, the entire pay-period of 1 or 2 weeks is run on a cumulative basis, and bonus is computed on the average efficiency attained for the period.

GROUP ROUTING

PART NAME Piston MODELS 6-44 PART NO. C-3738

DATE 2-25-22 ISSUE NO. 37 NO. OF SHEETS 1 SHEET NO. BOOK NO.

DEPT.	DIV'N	GROUP	OPERATION	OPN. NO.	DETAIL EQUIP. NO.	DETAIL REQ. NO.	DETAIL TIME	REGISTER NUMBER	S. T. HOURS	HOURLY PROD.
571	A	1	Rough and finish C-bore, face and chamfer open end, rough and finish turn ring grooves and face top, turn O.D., drill piston pin oil holes, drill and ream piston pin hole, saw 2 horizontal slots, blow and place in grinders tray and set up machine.					X-1217	.059	16.5
			Rough and finish C-bore, face and chamfer open end	5	L7-4	R-1843	.013			
			Rough turn ring grooves and rough face top	10	AS-1-6					
					-7	R-1457	.006			
			Finish turn ring grooves and finish face top and turn O.D.	15	AS-2-3	R-1466	.0097			
			Drill and ream piston pin hole	20	DS-2-7					
					-25	R-1416	.0143			
			Drill piston pin oil holes (2)	25	DS-24	R-994	.0067			
			Saw 2 horizontal slots	30	HA-2-4	R-1786	.01			
			Blow and place in grinders tray	35		R-1844	.002			
			Set up machine	40		R-1804	.0078			
		2	Grind ring and diameter and grind body diameter elliptical					X-1218	.055	18.2
			Grind ring and diameter	45		R-1770	.0186			
			Grind body diameter elliptical	50		R-1768	.0365			
		3	Burr file horizontal slots, open and scrape inside of slots and blow and cut ring grooves deeper.					X-1219	.015	77.
			Burr file horizontal slots and open and, scrape inside of horizontal slots and blow	55		R-1845	.0125			
			Cut ring grooves deeper and remove fillet	60		R-1817	.0006			
			Credit on C.P. 152 by Material Dept. for number of pieces passed as good product at end of Dept. 571.							
			Credit on C.P. 148 for vendor scrap. 8% machine scrap allowed.							
			Effective Feb. 24, 1922.							

ISSUED TO BOOK NOS. 1-2-3-4-5-6-7-13-14-17-25-33-45 KIND OF MATERIAL lynite

FIG. 2—TYPICAL ROUTING SHEET GIVING THE STANDARD TIME AND THE TOOL-GROUP STANDARDS FOR THE DETAIL OPERATIONS ON A PARTICULAR PART

Obviously, for a given product, the fewer actual man-hours there are in a group, the higher the efficiency of production will be and the greater the per cent of bonus earnings. The incentive is, therefore, to obtain a high rate of production per man per hour consistent with good quality of product.

Each member of a group receives the same percentage of bonus at the end of the pay period, but it is computed on his wage earned while assigned to the group; and the total amount earned by each worker will, therefore, depend on his hourly base-rate and number of hours worked. No job tickets are used. A shop timekeeper will handle from 300 to as high as 600 group workers. A list of employes is tabulated daily for each group, and elapsed time is taken at the end of the day from the entrance time-clock. If a worker is transferred out of or into a group, a transfer slip noting the time is recorded by the timekeeper and the elapsed hours are charged accordingly. Standard-hours credit is obtained from the finished inspection reports of quantities, multiplied by the group standard-time. The forms used in this connection are reproduced in Fig. 3. The method of tabulating group records daily in the time office, and of computing bonus, is indicated by the form shown in Fig. 4.

A group may produce various parts having different time-standards, and therefore different direct-labor costs and, since we use no individual elapsed-time job-tickets, labor costs are computed from the group cost per standard-hour. For given base-rates, the labor cost will be constant per standard-hour and per piece for all efficiencies above 100 per cent. If the average efficiency falls to 90 per cent, the labor cost will increase less than 2 per cent.

In the above discussion, we have assumed that the production line has its operations arranged in sequence. Such lines are, of course, ideal for the grouping of workers. But in the class of factories we are considering, there may be anywhere from 5 to 40 per cent of the direct-labor operations that cannot be arranged progressively. Such departments usually consist of miscellaneous small machines, automatics, grinders, or minor bench operations. However, it has been our practice up to the present to group such departments according to similar operations, being careful to retain the personal interest of the worker in the group production. Credit is given only for finished parts as they leave the department. So far, such grouping has been entirely successful. If there were any special departments of work in

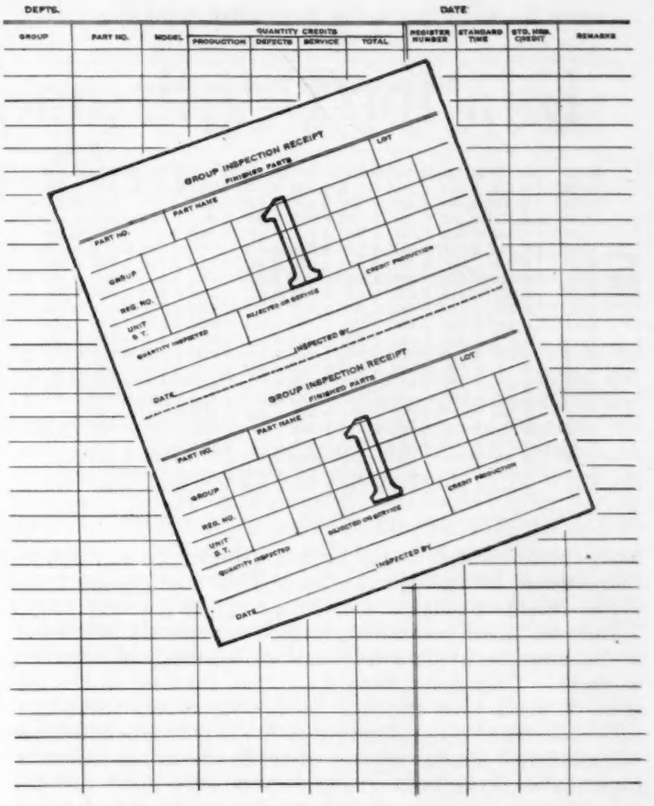


FIG. 3—FORMS EMPLOYED IN CONNECTION WITH THE SECURING OF THE STANDARD-HOURS CREDIT FOR A GROUP OF OPERATORS

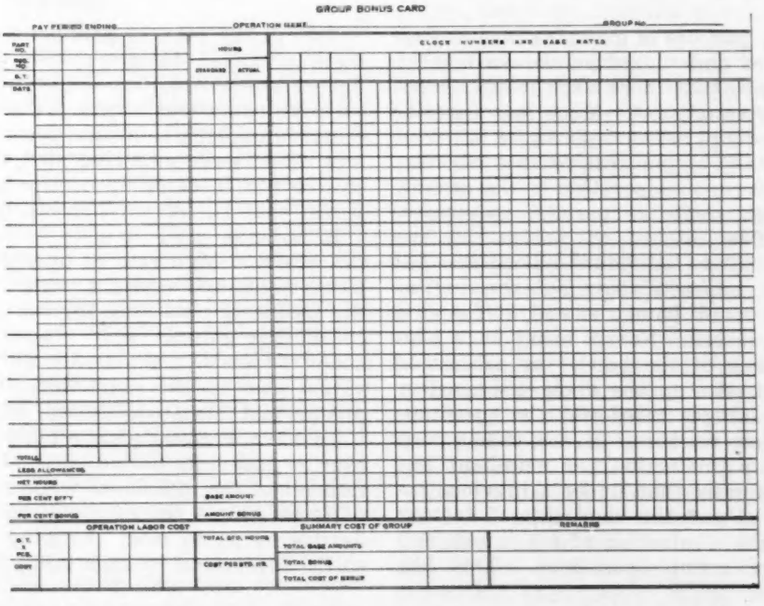


FIG. 4—CARD ON WHICH THE GROUP RECORDS ARE TABULATED DAILY IN THE TIME OFFICE AND THE BONUS COMPUTED

TABLE 3—WAGE TABLE FOR PRODUCTIVE WORKERS; COMPUTED FROM TABLE 1

Employe's Hourly Base Rate	Hourly Earnings at 85 to 130 Per Cent Efficiency									
	85	90	95	100	105	110	115	120	125	130
$0.25	$0.268	$0.275	$0.288	$0.300	$0.315	$0.330	$0.345	$0.360	$0.375	$0.390
0.26	0.278	0.286	0.299	0.312	0.328	0.343	0.359	0.374	0.390	0.406
0.28	0.300	0.308	0.322	0.336	0.353	0.370	0.386	0.403	0.420	0.437
0.30	0.321	0.330	0.345	0.360	0.378	0.396	0.414	0.432	0.450	0.468
0.32	0.342	0.352	0.368	0.384	0.403	0.422	0.442	0.461	0.480	0.499
0.34	0.364	0.374	0.391	0.408	0.428	0.449	0.469	0.490	0.510	0.530
0.35	0.375	0.385	0.403	0.420	0.441	0.462	0.483	0.504	0.525	0.546
0.36	0.385	0.396	0.414	0.432	0.454	0.475	0.497	0.518	0.540	0.562
0.38	0.407	0.418	0.437	0.456	0.479	0.502	0.524	0.547	0.570	0.593
0.40	0.428	0.440	0.460	0.480	0.504	0.528	0.552	0.576	0.600	0.624
0.42	0.449	0.462	0.483	0.504	0.529	0.554	0.580	0.605	0.630	0.655
0.44	0.471	0.484	0.506	0.528	0.554	0.581	0.607	0.634	0.660	0.686
0.45	0.482	0.495	0.518	0.540	0.567	0.594	0.621	0.648	0.675	0.702
0.46	0.492	0.506	0.529	0.552	0.580	0.607	0.635	0.662	0.690	0.718
0.48	0.514	0.528	0.552	0.576	0.605	0.634	0.662	0.691	0.720	0.749
0.50	0.535	0.550	0.575	0.600	0.630	0.660	0.690	0.720	0.750	0.780
0.52	0.556	0.572	0.598	0.624	0.655	0.686	0.718	0.749	0.780	0.811
0.54	0.578	0.594	0.621	0.648	0.680	0.713	0.745	0.778	0.810	0.842
0.55	0.589	0.605	0.633	0.660	0.693	0.726	0.759	0.792	0.825	0.858
0.56	0.599	0.616	0.644	0.672	0.706	0.739	0.773	0.806	0.840	0.874
0.58	0.621	0.638	0.667	0.696	0.731	0.766	0.800	0.835	0.870	0.905
0.60	0.642	0.660	0.690	0.720	0.756	0.792	0.828	0.864	0.900	0.936

such factories that could not be grouped to advantage, there would be no objection to leaving them on an individual-incentive basis or even on a straight day-rate basis.

Indirect labor has been grouped extensively wherever a "community of interest" can be maintained between workers. They must have a common interest in the results of their own efforts. Storeroom labor, unloading materials from cars, boxing and loading automobiles for shipment and similar classes of work where the effort is measurable, have been grouped with very good results.

In summarizing our experience during the past 4 years with the group-bonus plan, it should be borne in mind that the particular feature involved is the principle of grouping employes and not the wage-incentive table that happened to be selected. Very likely any one of several incentive plans could have been used in connection with grouping and have produced satisfactory results. This one was selected because we thought it would be more adaptable to changing factory conditions than a system of fixed-group piece-rates having their value in dollars rather than in time. It also offers the same incentive to high production as could be obtained from piece rates.

Although the primary purpose of grouping was to simplify the factory system and to reduce the amount of clerical detail, it developed that there were many advantages from an operating standpoint. Much less material is tied-up in process, and full advantage can be taken of the mechanical arrangement of progressive lines whereby parts are made to flow from one operation to the next in a steady stream through single or parallel operations. Under the continuous-flow method of production, the checking of quantities after individual operations becomes difficult, if not impracticable; so, the logical method seems to be to count from the end of the line and credit groups of workers. Nearly every such production line has its "neck of the bottle" or several of them. If these can be speeded up, the whole line benefits.

One of the early advantages noted was the speeding-up, or elimination, of slow workers. Hence, it has been our experience that more production per man-hour has been obtained under grouping than under a previous individual-incentive plan. From the viewpoint of factory operation it has meant the elimination of job tickets and elapsed-time records for group operations. This has meant saving in clerical detail. It has added to productive time, because there are always some delays if employes are required to keep count on quantities, obtain job tickets or furnish information to shop checkers.

Conditions Favorable to Grouping

It may be taken for granted that factory workers cannot be grouped indiscriminately, or under all sorts of conditions, with any hope of success. The individual incentive must not be lost sight of. If a group worker feels that others will carry his burden whether he exerts himself or not, there is no incentive for him to put forth his best efforts. Consequently, conditions must be such as will enable the individual to realize that the results are going to depend on his own interest in the work. Progressive lines of sequence operations, grouped according to major operations, are ideal for this purpose. On the other hand, where operations are not progressive but have a community interest, equally good results are obtained. As an example, a group of 20 men snagging miscellaneous castings on an average tonnage basis gave a remarkable performance because they had one interest, the output in pounds of castings per man per hour. Storeroom attendants, caring for and delivering materials, were successfully grouped by placing the incentive on the basis of hours time per 100 units produced by the factory.

The essentials for grouping are that workers must be placed close together as a team, and also be able to see the results of their own efforts. Operations should be largely repetitive. The results of each day's group-work should be made known as early as possible the following day to maintain the individual interest. The best results are obtained by tabulating this information cumulatively from day to day over a pay period of 1 to 2 weeks. It is essential that the computation of earnings be made so simple that the average factory worker can figure out his own earnings quickly, just as he would on straight day's-work. For this purpose a wage table, such as Table 3 but more in detail, is posted in the shop. Most employes use a copy of the bonus scale such as is shown in Table 1, and compute earnings directly from the average group efficiency.

Summary

The group plan is primarily applicable to repetitive work arranged in progressive production lines of sequence operations, but it is also applicable to non-progressive operations where the individual interest can be centered on definite results per man-hour of labor. Like any other wage-incentive plan, it has for its main objective the speeding-up of the production rate per employe; that is, *intensive production.*

It is being used because it simplifies the factory detail where a high rate of production is desired on repetitive work, and where a wage incentive is employed as the means of obtaining intensive production. Its application has gained favor among factory executives because it speeds-up production as compared with the individual plan on the same work, ties up less material in process and minimizes clerical detail in the factory.

To have a group plan remain in successful operation over an indefinite period, it is necessary to maintain the interest of the individual worker in the group effort. It must be simple for computation of individual earnings, flexible so as to meet changing factory conditions and easily understood by each group member.

The Control of Operating Tool and Supply Cost

By F. A. MANCE[1] DETROIT PRODUCTION MEETING PAPER

BY allotting to each department the precise amounts of tools and supplies that are to be used for each operation a control is established throughout the factory that must accord with the number of cars built. Over a period of 4 years a system of this kind in the plant cited has shown a decrease in the cost of renewals of 71 per cent. The methods of procedure are given for instituting such a system and for charging off the costs of tools used in one department when the tools are "salvaged" from other departments. A preferential list of sources of supply is recommended, which should contain the names of firms supplying the tools that have given the most satisfactory results in actual use.

A CONTROL over expenditures for perishable tools and operating supplies has been successfully established by rating or allotting the amounts to be used by each department for every operation. By this allotment we are able to regulate the amounts of tools and supplies that are used throughout the factories in accordance with the number of cars built. Each item is priced and extended, and the aggregate is totaled. The total represents the amount of money allotted to the department to produce a definite number of cars in a given period of time. In figuring departmental percentages of efficiency this amount is included with labor and production.

This system has been in operation for nearly 4 years and during this time the cost of renewals of perishable tools has decreased approximately 71 per cent. It is admitted, of course, that some of this reduction was due to a decrease in the cost of the tools.

To anyone interested in adopting this method we would suggest the following procedure: Start with the selection of a competent and aggressive tool-trouble man for making the survey. Provide work sheets on which to list the department, the part number, the operation, the description of both tools and supplies and the amounts allotted for any given number of cars per month. The allotment can be based on the actual withdrawals over a given previous period, after eliminating the waste on account of abuse and improper methods of use where instances of this kind are found. By this we mean that the allotment should be based on the amounts that should be used. These work-sheets are afterward to be used in making up a standard form in quadruplicate. The standard form should contain additional information as to price, the extensions and the total, as previously explained. The standard allotment-sheets when completed are distributed to the foreman interested, the tool or supply stores, the superintendent of the plant and the supervisor of tools or the methods and standardization department.

Deliveries from stores are made upon approved requisitions, provided the article wanted has been previously allotted and appears on the allotment sheets. Departmental operating tool and supply reports are made up for every 10-day period by the supervisor of stores and show the amounts withdrawn during that period. They are then forwarded to the accounting department for pricing. From copies of these sheets each foreman is checked as to the amount he is running over or under his allotment and he is allowed to see these copies during each period in order to acquaint himself with the cost of the articles that he uses. It is amusing at times to listen to the protests registered by the foremen against the prices paid for some of the more expensive items, but it gives one the satisfaction of knowing that he is vitally interested at all times in keeping down his operating costs.

Salvaged or restored tools are carried in stock and are given out on requisition in the same manner as new tools, with the exception that the requisition is stamped with the word "salvage" in red ink. This signifies that there is to be no charge against the department drawing out this material. When a drill has become too short for use in one department it is turned in as "salvage" and reconditioned for use in another department. This is also done with cutters, reamers and grinding wheels. The allotment sheets show where the salvage should be used and each item so salvaged is listed "no charge."

The department first drawing out a new tool bears the entire cost of the tool. In the case of special tools made to conform to a blueprint, the tool-design department notes on the tracings whether the tools can be salvaged from either a standard or another special tool of similar design, and these tools are also listed "no charge" on the allotment sheets.

The tool-salvage department keeps in touch daily with the various department foremen, advising them of available salvage tools, especially when these are in addition to the tools specified on the allotment sheets. The use of salvage tools so far this year has amounted to $1 per car. This sum represents what the cost would have been had we purchased new tools. The amount of money spent to recondition the tools amounted to 30 per cent of the original cost. This 30 per cent is charged off as expense, and is pro-rated over the entire portion of the plant that is benefited by the use of the tools.

One important factor that has a direct bearing on tool cost is the listing of sources of supply. This list is made up by the methods and standardization department and contains the names of the firms whose tools have given us the best results in actual use over a period of from 1 to 4 years.

The smallest possible number of firms is listed that is consistent with good business, and the relative merits of the firms can be determined from the order of their listing. When only one firm is given, no other satisfactory source has been found. When several are grouped, each firm is considered as good as any other of that particular group. Changes and additions are made by the methods and standardization department only when satisfactory

[1] Production department, Studebaker Corporation of America, Detroit.

results have been obtained from comparative tests and from use. This approved list of sources is distributed to the different purchasing departments for reference in buying.

So vast an amount of operating supplies, such as grease, solder, paints, oils, enamels, and gasoline, is used in the automotive industry that no effort should be spared to eliminate waste. On these particular items our allotment has accomplished a considerable saving in the quantities used from year to year.

In connection with the allotment of tools, we have begun recently a study of the life of perishable tools. Among drills this study consists, first, of a microscopic inspection of the steel structure to determine good heat-treatment. Various sizes of drill that show proper heat-treatment are selected and used for drilling steel forgings of a given Brinell hardness. The actual amount of metal removed per grinding is noted, and the drills are compared as regards life per inch of flute length and number of feet of metal removed.

Several charts have been made and an average has been computed for use as a reference in determining the actual life of drills for a given number of feet of steel of a certain range of Brinell hardness.

In all non-productive departments the expenditures for operating supplies are controlled by a fixed sum or budget and are based on car production in the same manner as in the productive departments.

The tool and supply stores are controlled by requisitioning only such items as appear on the allotment sheets. All purchase requisitions from stores must state the amount used in the previous 3 months, the approximate total cost, the amount in stock at all plants and whether other material can be substituted or salvaged. It is essential that the man who supervises the allotment should check all requisitions before they are approved.

OCTOBER COUNCIL MEETING

THE meeting of the Council held in Detroit on Oct. 25 was attended by President Bachman, Past-President Beecroft, Vice-Presidents Watts and Young, Councilors Brush, Crane, Strickland and Scott; and W. A. Chryst and A. J. Scaife.

A preliminary financial report for the fiscal year ended Sept. 30 last was considered. This showed that the deficit for the year was about $15,000. This deficit is approximately half as great as that specified in the budget. The result of the year's operations was considered very satisfactory under the circumstances, the deficit being practically equal to additional expense incurred during the last fiscal year for activities consequent upon the enlarged scope of the Society's work. The total income for the last fiscal year was $248,000 this being $16,000 more than contemplated by the budget for the period, and $5,200 more than the actual income during the fiscal year ended Sept. 30, 1921. The net assets of the Society on Sept. 30 amounted to $122,000.

It was decided to issue a 1923 Roster of the Society in the same form as heretofore, copies thereof being furnished without charge to members requesting them.

A long discussion was had of proposed programs for the 1923 Annual and Summer Meetings of the Society. The chief difficulty in scheduling sessions is to allow enough time for discussion and still have enough of the available desirable papers presented without holding too many sessions at a meeting. It was decided definitely to discontinue the Carnival that has been held in recent years in connection with the Annual Meeting. The sentiment at the Council meeting was not in favor of a boat trip for the Summer Meeting.

Thirty-eight applications for individual membership were approved. The following transfers in grade of membership were approved: Junior to Member, R. E. Bissell, F. L. Creager; Associate to Member, W. J. Bryan, Joseph V. Petrelli, E. Burnell and Thomas A. Clark.

It was reported that up to Oct. 24, 1922, 611 applications for membership had been received during 1922, as compared with 625 received during the first 10 months of 1921, and 1033 during the first 10 months of 1920. A material gain in membership will be shown at the beginning of 1923. At the present time the net membership of the Society is about the same as it was at the corresponding period of 1921, the Council having recently dropped a large number of members for non-payment of dues.

The following subjects were assigned to the respective Divisions of the Standards Committee:

Engine Testing Forms—Motorboat Division

Motorboat Trial Performance Forms—Motorboat Division

Isolated Electric Lighting-Plant Storage-Battery Rating—Storage Battery Division (reassignment)

It was decided not to assign the subject of engine-bearing oil-grooves for consideration in connection with standardization.

THE PRODUCTION MEETING

(*Concluded from p.* 390)

members and the service of its local office. The meetings, visits and dinner were conducted punctually and smoothly because of the whole-hearted support of the entire group of members actively interested in the arrangements.

1923 Production Meeting

There will be a 1923 Production Meeting of the Society; that is a certainty. The spontaneous reception and representative attendance given the Detroit Meeting assure the closest attention being given to the production phases of the industry by the Society at National and Sections Meetings. Out of this new activity will come a better understanding between the man who designs and the man who produces. This can only mean a further strengthening of our industry. No one who attended the Detroit Production Meeting can have failed to secure much of value from it.

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Experience Notes from a Production Notebook

By H. J. Crain[1] and J. Brodie[1]

Detroit Production Meeting Paper *Illustrated with* Photographs and Drawings

While investigating the sources and causes of noise in automobiles during an extensive connection with one of the largest automobile companies, the authors recorded their experiences in the shop in the form of notes. Some of these are offered with a view to stimulating the discussion of the subject and with the hope that additional information will be brought out by an exchange of ideas, particularly on the problem of eliminating gear-noises. In many cases they found that noise was caused by failure to allow sufficient clearance for an adequate oil-film. And it was noted frequently that when one noise had been located and silenced another appeared that had not been apparent before. The topics that have been considered include the running-in of brake-bands, engine knocks, oil-pump gear-noise and that of gears in general, the clearances of ball bearings, backlash, and rear-axle bevel-gears.

THE experiences recorded in this paper have not been selected in accordance with a specific plan. No attempt has been made to cover any particular subject fully or to arrange the different descriptions with regard to a related sequence. It is possible that some of the experiences have been encountered or the methods have been used by other production executives but we believe that knowledge of these methods is not general and that it will be of interest to factory men. We hope that the discussion will bring out additional information on some of the matters treated, particularly on the perplexing problem of eliminating gear noise.

Running-in Brake-Bands

The increasing congestion on city streets and the seriousness of the automobile accident and collision situation should be convincing evidence of the need of proper adjustment of motor-car brakes. It would seem important that cars should be shipped from the factories with the brakes seated, and adjusted to overcome the rapid wear that usually occurs in driving the first few hundred miles. This rapid wear is caused by the ironing or smoothing of the brake-lining surface until the high spots have been worn down to the level of the rest of the lining face. It may be due also to slight imperfections in the contour of the brake-band. Figs. 1 and 2 illustrate two motor-driven machines designed and built by the Packard company for the purpose of running-in brake-bands. The machine shown in Fig. 1 handles the internal or expanding brake and that in Fig. 2 the external or contracting brake. The brake-drums rotate at a speed of approximately 1000 r.p.m. in both cases. The drums are cooled by water, circulated about the peripheries of the drums, so that the temperatures are never excessive. Pressure is exerted on the brake-bands by a weighted lever, which can be seen clearly in Fig. 1, the weight being adjusted so that the pressure is only great enough to assure a full bearing of the band on the drum.

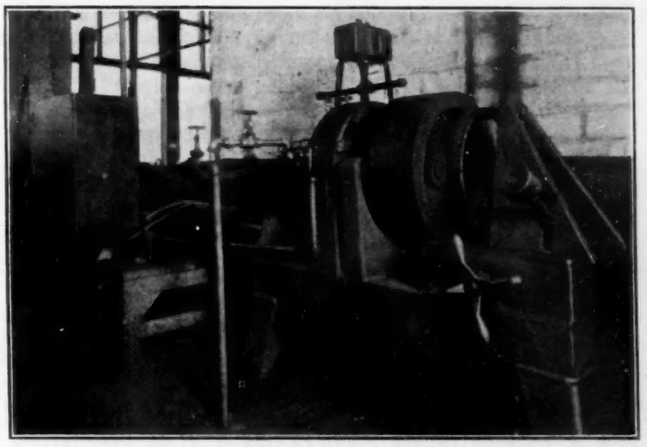

Fig. 1—Motor-Driven Machine for Running-In Internal Brake-Band Assemblies

Each band is run for about 1 min. The two machines are located so that the same operator can handle both; the band of one being run-in while the operator is loading the other. It will be found that the bands acquire a polished surface on these machines, and that the irregularities sometimes existing around the rivet-holes and throughout the lining surface are smoothed-out. By taking this precaution at the factory the maximum brake efficiency is attained at the beginning of operation of the car, and the adjustments usually required in a new car after a few days of service are unnecessary.

The importance of accuracy in grinding a piston skirt is recognized by all production men. The center, shown in Fig. 3, was originally used in the Packard shops for centering and driving the pistons during the external grinding operation. Excessive wear of the surface *a*

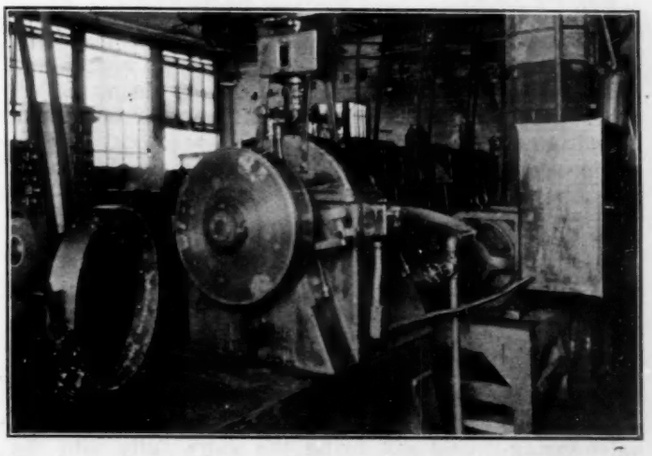

Fig. 2—A Special Machine Developed by the Packard Company for Running-In External Brake-Band Assemblies

[1] Production department, Packard Motor Car Co., Detroit.

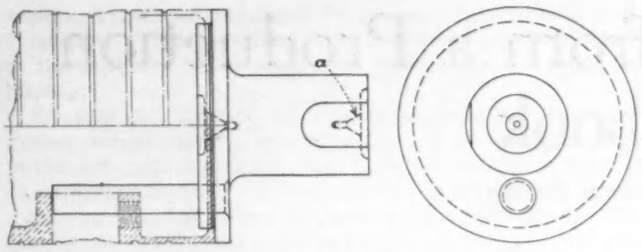

FIG. 3—FORM OF CENTER ORIGINALLY USED IN THE PACKARD SHOPS FOR CENTERING AND DRIVING THE PISTONS DURING THE EXTERNAL GRINDING OPERATION

necessitated the frequent replacement of this center and demanded constant supervision by the foreman in order that the work should not be spoiled by continuing the use of a center that had passed the permissible stage of wear. The study given to this small but puzzling problem has resulted in the adoption of the center shown in Fig. 4. In this instance the thrust and driving load are taken by a taper-roller bearing of heavy load-capacity, the wear is distributed over a very large surface, lubrication is easily maintained and the life of the center is greatly prolonged. This design has proved very successful, and, no doubt, other tool designers could apply it to advantage. This particular center is used with Brown & Sharpe Nos. 12 and 14 external grinding-machines.

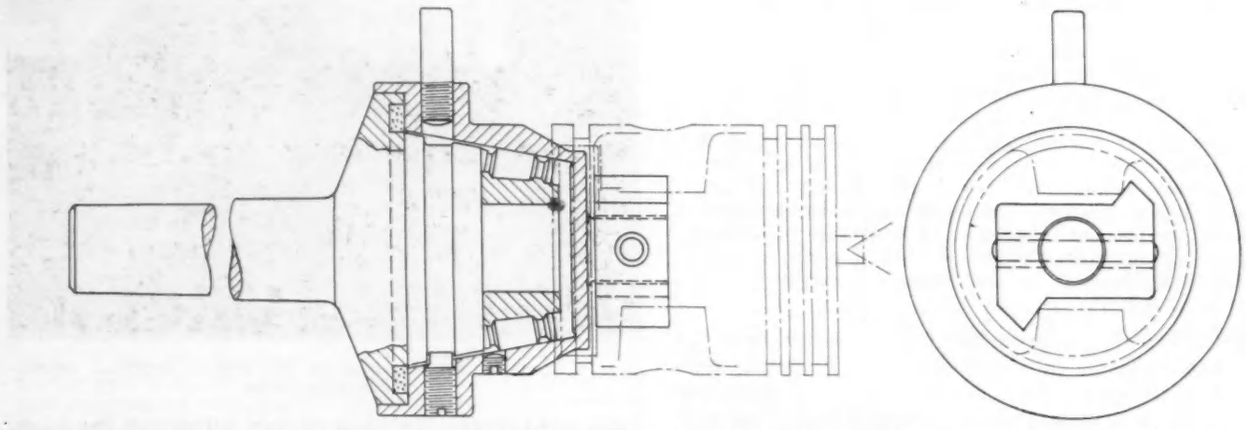

FIG. 4—A NEWER FORM OF CENTER IN WHICH THE THRUST AND THE DRIVING LOAD ARE TAKEN BY A TAPER ROLLER BEARING

A PUZZLING ENGINE KNOCK

All production and inspection departments have had the displeasure of running down peculiar engine knocks. The following note from Packard experience may shed some light on this trouble. A few years ago when a new model was started through the Packard shops the engines of the first run received at the test-stands were all found to have a perceptible piston slap or knock. Numerous remedies were tried but eventually the real cause of the noise was found largely through accident.

The click came only at the time of the explosion. Investigation revealed the fact that the valves were not centering properly in the conical surface in the cylinder. The condition is shown in exaggerated form in Fig. 5. It was found that the tool used to form the valve-seat was centered by a spindle inserted in the valve-stem guide. This spindle was too much undersize and allowed the tool to float just enough to throw the conical seat out of alignment with the valve-stem guide. As a result, the valve-spring would not bring the valve fully into the seat; the valve would hang on one side of the valve-seat until the explosion snapped the valve into the seat with a very noticeable click. Of course this part of the noise was then obviated without difficulty.

But this correction did not stop all the noise. A more annoying knock was eventually found to come from the piston-rings, which were of the diagonal-cut type with a slight clearance between the ends. As the explosion-pressure reached the rings they were compressed and their ends snapped. Filing several grooves in the upper edge of the top ring would let part of the explosion in behind the ring, expand the ring and overcome the slap until the grooves had filled with carbon. A ring that slaps in the manner described generally shows bright polished ends. When the proper end-clearance had been determined by experiment, the noise ceased. The rings in general use to-day have overlapping joints; the end clearance can be very large and, of course, this trouble is not encountered.

OIL-PUMP GEAR NOISE

Oil is circulated in Packard engines by a gear-pump similar to that illustrated in Fig. 6. When this particular design was first adopted it was found to produce a very irritating noise, which sounded like the blades of a fan striking a sheet of paper. Naturally this was

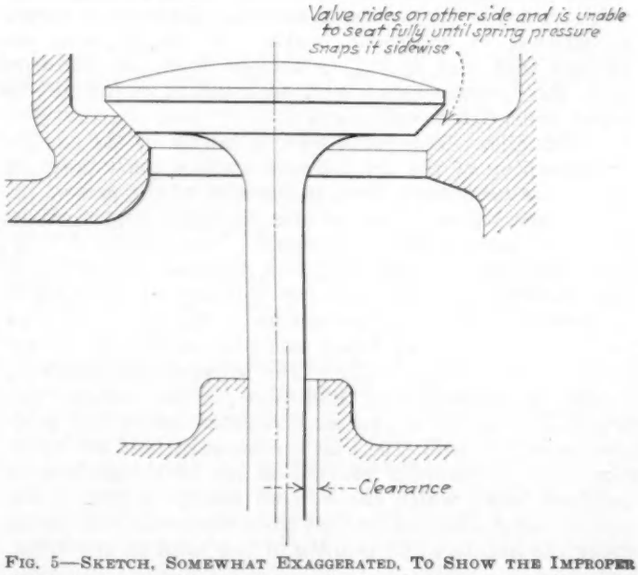

FIG. 5—SKETCH, SOMEWHAT EXAGGERATED, TO SHOW THE IMPROPER CENTERING OF THE VALVES IN THE CONICAL SURFACE IN THE CYLINDER

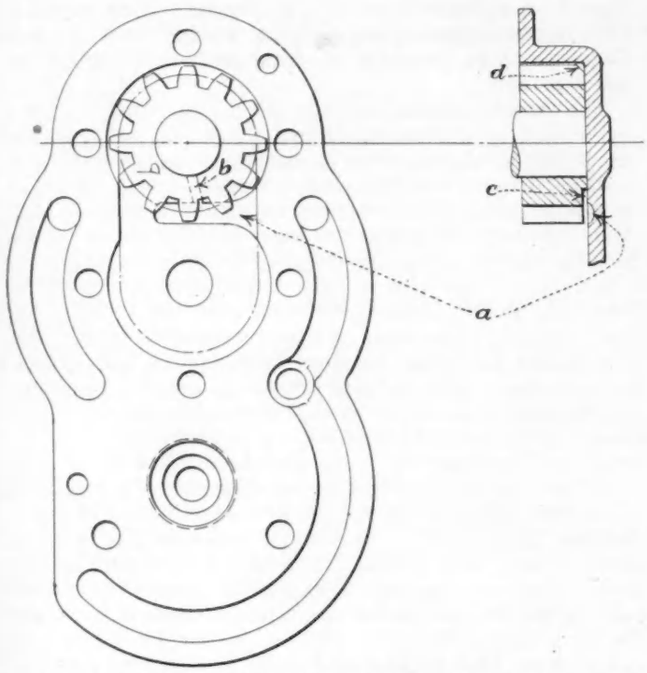

FIG. 6—THE GEAR-PUMP THAT CIRCULATES THE LUBRICATING OIL IN THE PACKARD ENGINE

attributed to imperfections in the gears. In the experiments made to abate this nuisance tooth-forms and pressure-angles were varied, and helical and herring-bone gears were fitted, but the clatter persisted. All degrees of backlash were tried but without avail. It was noticed that a certain run of pumps were more quiet than the others. These were inspected carefully to ascertain what variation was responsible for the lessening of the noise. The only difference found was a slight relief on the lower face of the idler gear. At *a*, in Fig. 6, is shown a feeder channel that is cut in the base of the pump for the purpose of carrying oil to the idler bearing through the cross channel *b*. Note that this channel is open to the pressure side of the pump but ends at the point where tooth contact ceases on the suction side. It was found that when the tooth corners were beveled, as shown at *c*, the noise was reduced. The possible effect of these changes was the basis of a careful study, which resulted in the discovery of the real source of the noise. Both these schemes eliminated the sharp cutting off of the oil stream that would naturally attempt to escape at *d* from the pressure side to the suction side of the pump. By relieving the pressure in the groove *a* the oil was not able to spurt against the tooth faces and rattle the unloaded idler gear in the backlash space in the driving gear. When, as an experiment, the groove *a* was filled with solder, the altered pump became quiet. The design of the pump was changed, the groove *a* was omitted, and no further trouble was experienced. This case is cited as an example because it indicates that gear noises are not always attributable to the gear-teeth themselves.

GEAR-NOISE INVESTIGATIONS

The production and engineering staffs of the Packard Motor Car Co. have been studying the matter of gear noises for many years. This work is still being carried on but no panacea has been definitely discovered for gear troubles. Each case seems to have its peculiarities and to require special modifications to become silent. Alterations that are effective in one case might not be effective in another that to all appearances is similar. It is more than likely that the study being given to gear-tooth wear and noise in our own and other industries will lead eventually to a better understanding of the fundamental causes of the difficulties. For the present, however, it is only possible for us to exchange experiences for the common benefit of those who are interested.

A large number of investigations made over a period of years have led us to believe that gear noise does not always originate in variations of the gears themselves. Such variations undoubtedly contribute to the gear growl or chatter but the noise can often be cured or dampened by an alteration of another part. We have concluded that the proper mounting of the gears on rigid shafts is a paramount requirement if noise is to be avoided. The most perfect tooth-forms, ground, shaped or milled, will not run quietly if they are carried on shafts that spring or are not in perfect alignment. It is assumed that this essential fact is recognized in the engineer's design of transmissions and axles. Given a properly designed mounting, it is the factory man's problem to reduce noise. The factors controlling it are largely independent of blueprints. Drawings can only give the characteristics of gear-teeth; the production man must see that the actual contours and allowable variations keep them within an acceptable range of quietness.

The front end of a typical transmission is shown in Fig. 7. Particular attention is directed to the mounting of the main drive-gear *a*, in which it will be seen that this is carried on a roller bearing, the inner race of which is formed by the shaft itself and the outer race is mounted in the transmission case. In the final inspection of a certain model at the Packard factory it was noticed that the degree of gear noise varied from very

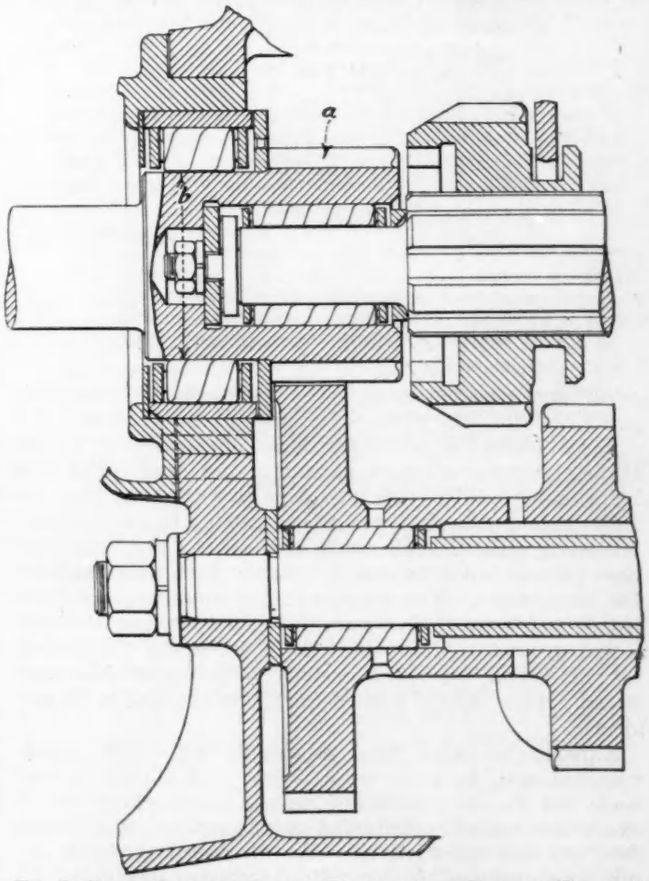

FIG. 7—FRONT END OF A TYPICAL TRANSMISSION SHOWING HOW THE MAIN DRIVE GEAR IS CARRIED ON A ROLLER BEARING

quiet to objectionably loud. Attention, naturally, was centered on the noisy gears. These were returned to the transmission department and torn down for careful inspection, adjustment and reassembling. Invariably the inspection revealed gears, bearings and shafts that were as near perfection as it seemed possible to approach. This led to the assembling of special gears in which perfection was carried to the utmost degree, a state far beyond that possible under even unreasonable inspection practice. But the noise, if anything, was worse.

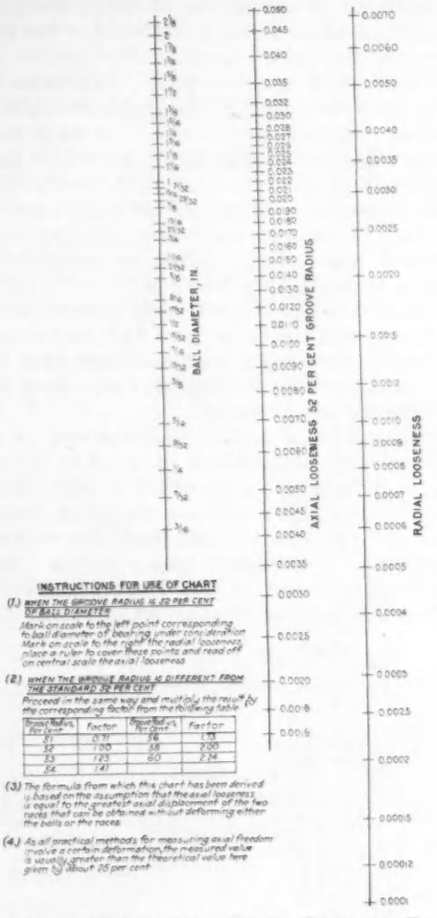
BALL DIAMETER, IN.
AXIAL LOOSENESS 52 PER CENT GROOVE RADIUS
RADIAL LOOSENESS
INSTRUCTIONS FOR USE OF CHART
(1.) WHEN THE GROOVE RADIUS IS 52 PER CENT OF BALL DIAMETER
Mark on scale to the left point corresponding to ball diameter of bearing under consideration. Mark on scale to the right the radial looseness, place a ruler to cover these points and read off on central scale the axial looseness
(2.) WHEN THE GROOVE RADIUS IS DIFFERENT FROM THE STANDARD 52 PER CENT
Proceed in the same way and multiply the result by the corresponding factor from the following table
Groove Radius, Per Cent | Factor | Groove Radius, Per Cent | Factor
51 | 0.79 | 56 | 1.73
52 | 1.00 | 58 | 2.00
53 | 1.23 | 60 | 2.34
54 | 1.41
(3.) The formula from which this chart has been derived is based on the assumption that the axial looseness is equal to the greatest axial displacement of the two races that can be obtained without deforming either the balls or the races
(4.) As all practical methods for measuring axial freedom involve a certain deformation, the measured value is usually greater than the theoretical value here given by about 25 per cent

FIG. 8—NOMOGRAPHIC CHART DESIGNED TO CONVERT THE DESIRED RADIAL CLEARANCE OF A BALL BEARING INTO THE EQUIVALENT AXIAL PLAY

It remained for us to tear down and to inspect several transmissions that were passed in the final car-test as being quiet. When this was done it was found that the roller race on the shafts had been ground to the low-limit diameter, while the roller race in the shells had invariably been ground to the largest or extreme high limit specified for these holes. For purposes of comparison, six noisy transmissions were then torn down, the bearing diameter *b* was ground approximately 0.001 in. under the former low limit and the transmissions were reassembled and tested. This change caused the noise practically to disappear.

Experiments were made repeatedly with noisy transmissions and, in every case, when this alteration was made and the bearing clearance was increased, the noise was either entirely eliminated or was reduced to a degree that was not objectionable. We concluded that this result was produced by providing sufficient space for an adequate oil-film. Further experiments along similar lines have substantiated this conclusion. This explanation seemed logical since we have always found such a clearance to be necessary in crankshaft and connecting-rod bearings.

The remedy seemed a simple one to supply but we were quite concerned about mounting a bearing under conditions that simulated those it would assume after several months' wear. We had always supposed that bearings of the anti-friction type must be mounted snugly. Before definitely adopting the new practice, wisdom demanded that we check the effect of the greater diametral clearance on the wear of the bearing. Transmissions were run under similar conditions with the standard or snug bearing and with the increased clearance. We found that the snug bearing wore rapidly during the early stages of the test and eventually reached the state of looseness with which the other bearing started. The loose bearing, on the contrary, practically retained its original clearance. We concluded that the wear of the snug bearing was accelerated because of the absence of an oil-film sufficient for complete lubrication. The loose bearing apparently accommodated an adequate oil-film and the wear was normal. The test was continued for some time and frequent examination showed that the snug or full-fitting bearing continued to wear faster than the loose one. This, we believe, is due to the heavy initial wear that breaks or distorts the ground surface instead of glazing it as seems to be the case when the bearing is assembled with a proper clearance at the start.

BALL-BEARING CLEARANCES

After the transmissions using the roller bearing had been changed to conform to the practice just described, it became apparent that the same gear-noise existed in the transmission used in one of the other Packard models which had the transmission gears mounted on ball bearings. We altered a few experimental ball bearings by deepening the grooves in the races to allow a minute clearance for the balls, instead of assembling them to a good rolling-fit or to the fit of the standard stock. This was done to provide oil clearance, as had been done with the roller bearings. When the loose ball bearings were substituted for the tight ones in noisy transmissions, our previous experience was repeated and the noise decreased. Careful experiments determined the desirable clearance to allow in the races for the reduction of noise, and the manufacturers of ball bearings agreed to supply bearings with various amounts of clearance so that we could determine the requirements of this work.

It was a comparatively easy matter to determine the clearance needed in the case of the roller bearings since diametral clearances could readily be measured. It was found to be difficult, however, to measure radial clearance in the ball-bearing races and we were forced temporarily to determine the degree of clearance by the end-play or axial looseness. A maker of ball bearings prepared the nomographic chart, reproduced in Fig. 8, for the purpose of converting a desired radial clearance into the equivalent axial play. This enabled us to order bearings with a selected radial clearance, which could be held uniform without demanding a finer tolerance from the bearing maker, and cured the gear noise most effectively.

We selected three groups of 10 bearings each and assembled the transmissions with the respective groups. Group No. 1 represented the average clearance in the bearings found in stock; No. 2 contained more radial clearance; and No. 3 still more or about 0.0006-in. radial clearance added to the old standard. These bearings

(*Concluded on p.* 437)

NATIONAL AERONAUTIC ASSOCIATION

THE National Aeronautic Association of the U. S. A. was formally organized in Detroit last month at the time of the holding of the Second National Aero Congress there and the national airplane races at Selfridge Field, Mount Clemens, Mich. Over 300 delegates from various parts of the Country attended the congress, and a large number of them took part in many arduous long committee and general sessions with the avowed determination to establish a national organization to foster the advance of the arts and sciences of aeronautics and allied interests. In this purpose they were successful.

The delegates represented groups of States divided into districts on the Government national defense basis. The congress was constituted of an unusually impressive body of men, and was conducted in a businesslike expeditious manner. The work of Lieut.-Col. Harold E. Hartney preliminary to the convention was very highly commended. The Detroit Aviation Society gave excellent cooperation in connection with the congress, as well as in the conduct of the races.

The following were elected as officers of the National Aeronautic Association for the ensuing year:

President, H. E. Coffin, Detroit
Vice-President, B. M. Mulvihill, Pittsburgh
Treasurer, B. F. Castle, New York City
Secretary, J. B. Coleman, Sioux City, Iowa

Governors

S. D. Waldon, Detroit
P. H. Adams, Boston
G. L. Cabot, Cambridge, Mass.
J. D. Larkin, Jr., Buffalo
M. J. Cleary, New York City
L. F. Sevier, Pittsburgh
R. F. Walters, Baltimore
A. S. Fliery, Birmingham, Ala.
F. H. Burgin, Atlanta, Ga.
G. L. Martin, Cleveland
D. M. Outcalt, Cincinnati
C. S. Rieman, Chicago
Ralph Cram, Davenport, Iowa
Howard Wehrle, Kansas City, Mo.
Edgar Tobin, San Antonio, Tex.
William Long, San Antonio, Tex.
P. G. Johnson, Seattle, Wash.
C. H. Messer, Spokane, Wash.

RACES

The winners of the races were as follows:

Curtiss Marine Flying Trophy—Navy T R 1 plane (Lieut. A. W. Gorton) Average speed, 112.65 m.p.h.

Detroit News Aerial Mail Trophy—Army Martin Transport (Lieut. E. H. Nelson) Average speed 105.1 m.p.h.

Aviation Country Club of Detroit Trophy—Army Honeymoon Express (Lieut. H. R. Harris) Average speed 135.1 m.p.h.

Liberty Engine Builders' Trophy—Army Lepere observation plane (Lieut. T. J. Koenig) Average speed 128.8 m.p.h.

Pulitzer Trophy—Army Curtiss high-speed pursuit plane (Lieut. R. L. Maughan) Average speed 206 m.p.h.

AIR INSTITUTE MEETING

An interesting session was held on the morning of Oct. 11 under the auspices of the Detroit Aviation Society. Prof. H. C. Sadler, of the University of Michigan, presided; the program having been arranged by a committee representing the National Advisory Committee for Aeronautics, the American Society of Mechanical Engineers, the Aeronautical Chamber of Commerce and the Society of Automotive Engineers. John R. Cautley took an unusually active part in the preparations for the meeting.

Dr. Joseph S. Ames, of the National Advisory Committee for Aeronautics, sent a communication on the Importance of Scientific Investigation in a General Aeronautical Program. L. E. Pierson, president of the Merchants Association of New York City, contributed remarks on Commercial Aviation and the Commercial Bank. C. F. Redden, president, Aeromarine Airways, had as a topic Flying-Boat Transportation. W. P. MacCracken, Jr., chairman of the Aviation Committee of the American Bar Association, gave a comprehensive outline of Aeronautical Legal Problems. Edmund Ely, president, National Aircraft Underwriters Association, spoke on the status of Aircraft Insurance. Prof. E. P. Warner, of Massachusetts Institute of Technology, prepared for the session a paper reviewing developments in Europe from the standpoint of Commercial Flying. C. G. Peterson presented an analysis of a proposed Contract Air-Mail Route between Chicago and New York City.

THE AIR MAIL

Col. Paul Henderson, second assistant postmaster general, in connection with The Air Mail said:

> Our service at present consists of a relay advance of mail from New York City across the Continent, and vice versa. That is to say, we do not take any particular mail for a complete trip across the Continent. We advance into Cleveland certain mail which misses the late night-trains out of New York City. We take from Cleveland into Chicago mail which, if we did not carry it, would go into Chicago on a train too late for delivery in the afternoon. This process is repeated in relays across the Continent with the net result that we advance approximately 12,000 lb. of first-class letter mail each day by some 3 or 4 hr. It should be noted that this may in certain instances mean a real advance of 15 to 18 hr., inasmuch as it may result in the delivery late in the evening of mail which might otherwise have not been delivered until the following morning.
>
> The planes that we are now using are remodelled DeHavilands, which we procure free of charge from the Army. As of this date, we have 70 such planes in flying condition. Twenty are in the air each day and about 24 are in process of being overhauled and rebuilt. Our engineers have found it necessary to make some 200 changes in the design of the plane to make it suitable for carrying mail.
>
> We use Liberty engines, also procured without charge from the Army. Our experience has shown us that at the end of 100 hr. flying service it is necessary to overhaul each engine. This we do at an average cost of about $250 per engine. At the end of 300 or 400 hr. flying service we overhaul the planes themselves. The major portion of this overhauling and rebuilding is done in our shops at Chicago, which are rather complete, approximately 100 persons being employed in them.
>
> From July 16, 1921, until Sept. 7, 1922, we flew approximately 2,000,000 miles without a fatal accident. During the fiscal year ended June 30, 1922, we maintained an efficiency of 94.39 per cent. This means that out of every 100 trips scheduled, 94.39 were finished on schedule time. Our records show that two-thirds of our trips were made in clear weather; one-third in foggy, cloudy or stormy weather.
>
> On Sept. 16 we finished 10 consecutive weeks of flying the entire Transcontinental route with 100 per cent efficiency; that is to say, during those weeks each of our scheduled trips was started and finished exactly on schedule time. It should be remembered that our daily route includes the crossing of three mountain ranges, the Alleghanies, the Rockies and the Sierras.
>
> There are employed in the Air Mail Service 390 people, of whom 39 are pilots. With three or four exceptions, our pilots are all "ex" Army or Navy flyers. They are exceptionally high-grade young men

and to them is due much credit for the success of the Air Mail Service.

Night Flying

It is obvious that to get from the airplane all that it may offer in the shape of postal service it will be necessary to fly at night. With this thought in mind, we have for the past 4 months conducted an intensive series of experiments on this subject. I think we have reached the stage where it is safe to conclude that it is entirely possible to fly at night. We expect within a few weeks to light, as an experiment, our Chicago field, and I am optimistic enough to predict that within 6 or 8 months we will be able to fly from Chicago to Cheyenne at night.

If we are successful in this, it will mean that we will be able to make a Transcontinental flight from New York City to San Francisco in one continuous movement, flying from New York to Chicago in the daytime, from Chicago to Cheyenne at night, and from Cheyenne to San Francisco during the early part of the second day. We should be able to establish and maintain a schedule of from 28 to 30 hr. between New York and San Francisco if this night-flying experiment proves out.

Our plan for night flying includes an emergency landing-field every 25 miles, with the proper field-lights, and a beacon-light visible for a distance in excess of 25 miles.

It is my personal opinion that within 2 or 3 years the Air Mail Service will have developed to a point where it will undoubtedly be thought wise to turn over the service to private contractors and make it nation-wide in scope, with higher postage than is now charged on ordinary letters.

The position of the Post Office Department in the matter of the Air Mail Service is that such information as we are able to develop and such experiments as we are able to follow through to a conclusion are for the benefit of the Country at large. If in our work we shall be able to add our share toward the prompt advancement of aeronautics, we will have done our duty.

Commercial Air-Transport

J. Rowland Bibbins, manager, department of transportation and communications of the Chamber of Commerce of the U. S. A., pointed out that

the American Nation is now spending for transportation about $100 per capita per year, far more than the whole prewar national debt, and nearly half the present national debt. If all business transactions could be done at one place and time, there would be no need for long-haul transport. But during the year, commerce requires a movement of 500,000,000,000 passenger-miles and 50,000,000,000 ton-miles on the railroads alone, neglecting entirely the enormous movement, yet unchartered, of 10,500,000 motor vehicles.

Outside of the actual transport cost, a great and unknown cost is the time element in the transaction of business, not only of personnel, but of mail, specie, securities, bank clearings and urgent merchandise, as well as less valuable and urgent freight and express. Every added hour or even minute in transit adds to the cost of doing business, in personnel, interest carrying charges, additional equipment needed in transport, and additional working capital assets of business. This conception is not visionary, it is an actuality, and has given rise to various methods and agencies for expediting business, for which service additional rates are paid, and paid gladly. Here is an open field, and a fair field, for air transport.

I think it is not an exaggeration to say that railroad investment other than main-line represents perhaps one-half of the total investment to-day, or $10,000,000,000. And we have estimated roughly a total investment in marine terminal facilities, other than railroad tide-water facilities, included above, of at least $1,000,000,000. No man knows how much of this tremendous investment could have been avoided by starting right, with a reasonable unification of effort and cooperative organization such as now have come about in the 50-odd railroad belt-line companies in the United States, as exemplified in the Railroad Terminal Association of St. Louis, and the plan now underway for the New York industrial district; but we can draw from past experience and see to it that in this great development that is to come, terminal duplication with its superburden of cost and inconvenience to the traveling public and traffic, should be reduced to the practical minimum. In fact, while it is a far look ahead, this may possibly become the key to maximum future success of air transport.

There is no profit in speculating what ultimate relation will be established and proved desirable between air transport and other transport agencies. We are learning rapidly, however, through the tragic experience of war, and bald necessity to-day, that our National transportation plant is in essence a single great problem and not an unrelated series of separate problems. And experience is rapidly being accumulated which will enable us to determine in the not distant future just what special field and radius of operation as well as economics will develop the best that lies in each form of transportation, rail, motor, trolley, canal and ultimately air.

Our department has estimated roughly the present-day investment in our national transportation system. It is $50,000,000,000, exceeding manufactures, mining and all others except agriculture. This investment has doubled within about 10 years, and previously within about 15 years. Of course, the recent great activity has been in highway and automobile transport. What of the future?

The basic rail-tonnage per capita has risen steadily to 25 tons in 1920, hardly without interruption. Railroad investment per ton carried has reduced through economic methods to practically a stable level of the last decade. Our population and tonnage demand will increase by a substantially known amount. During the next 20 years this combination of events will demand at least $25,000,000,000 new capital for transportation in all forms.

In the meantime probably half of the whole transportation plant representing the depreciable elements, will pass through the renewal cycle, a polite term for the scrap-heap, $25,000,000,000 more. A total of $50,000,000,000 in transportation funds will thus have to be raised and spent somehow. This is twice our national debt to-day. Looking backward as well as forward, the real potentiality of a new speed-service appears in its true light, for every additional increment in speed of transit will release just so much existing capital for non-speed service. I am no prophet, but the facts are irresistible.

Chassis Friction Losses

By E. H. Lockwood[1]

Metropolitan-New England Sections Paper *Illustrated with* Photographs and Charts

THE loss of power due to the friction of the various parts of the chassis has been carefully and elaborately investigated by a dynamometer, the dual purpose being the determination of the amount of internal frictional resistance of the front or rear wheels and the measurement of the power that can be delivered at the rear wheels with the concomitant rate of fuel-consumption.

The rolling-friction due to the resistance of the wheels as a whole is taken up first and afterward the separate resistances of the tires, bearings and transmission are studied under varying conditions of inflation-pressure and load. The five frictional resistances that were chosen as giving the most useful information are those of the front tires, the rear tires, the front bearings, the rear bearings and the engine.

Among other topics considered are the ratio of the total friction less the weight of the vehicle; a comparison of the resistance of passenger cars and trucks, of solid tires with pneumatic and of fabric tires with cord; the ratio of tire-friction to bearing friction; the rules for determining the total friction of the chassis; the effect of variations in the load and the inflation-pressure on the rolling-resistance of pneumatic tires; the development of resistance formulas for fabric and cord tires; a comparison of the wear of pneumatic tires with that of solid tires under the same load, the effects due to variations of the speed and to heavy-wall tubes, roughness of the tread, non-skid surfaces, large and small sizes, age and duration of wear; the development of heat in the tires; and the influence on friction of the rise of temperature of the gearbox lubricant.

The apparent decrease of friction during the last few years and the uniformity of the products of certain manufacturers are noted. The results of the tests are shown in detail by numerous charts and tables.

THE dynamometer drums consisted of metal-shrouded paper cylinders, mounted on a heavy shaft hung from the basement ceiling on ring-oiled babbitt-bearing hangers. The tops of the drums projected slightly above the main floor through openings in the concrete. The diameter was about 67 in.; the faces were 15 in.; and the overall width was 71 in.

Dynamometer measurements of a chassis were made by placing one pair of wheels on the tops of the drums, so that the wheels and drums revolved in rolling contact. Two distinct ends were sought: (*a*) the determination of the amount of internal frictional resistance of the front or the rear wheels; (*b*) the measurement of the power that the engine can deliver at the rear wheels, and the rate of fuel-consumption.

The power to drive the drums was obtained from a 15-hp. variable-speed electric motor which was belted to the drum-shaft. The power delivered to the drum-shaft by the engine of the car was measured on a prony-brake pulley of 100-hp. capacity. The drum-shaft rotation was measured by two ratchet counters and by an electric tachometer reading in revolutions per minute. All the apparatus could be observed and controlled from an operator's table on the main floor near the drums.

Fig. 1 is a view of the dynamometer from the basement, showing the driving motor and the prony brake. Fig. 2 shows the operator's table with the starting rheostat and meters for the electric motor, drum tachometer, revolution counters and scales for weighing the prony-brake load. The entire apparatus is shown in vertical section in Fig. 3 to which reference may be made by the following letters:

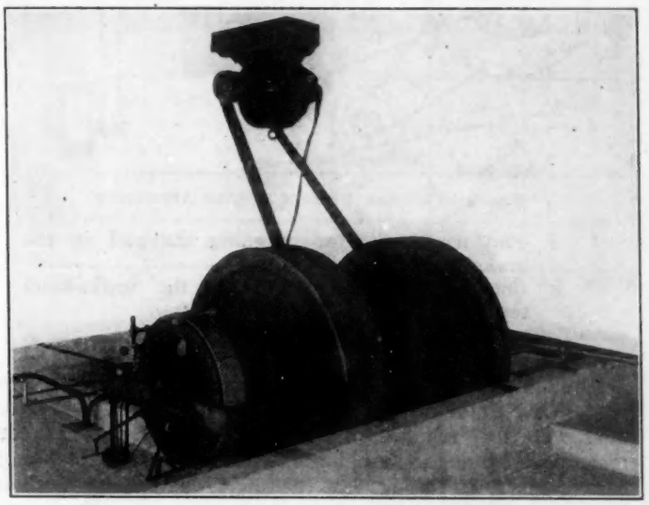

Fig. 1—View of the Dynamometer from the Basement Showing the Prony Brake

a, *a*, Main drums
b 15-hp. variable-speed electric-motor belted to the drums
c Prony-brake pulley on the main shaft
d Electric tachometer, geared to the main shaft
e Indicating dial of the electric tachometer
f, *g* Direct-current meters for the 15-hp. motor
h Starting rheostat for the 15-hp. motor

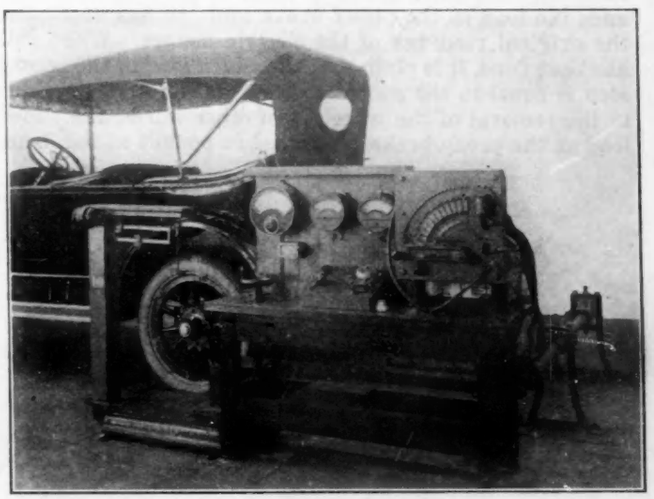

Fig. 2—Operator's Table

[1] M.S.A.E.—Assistant professor of mechanical engineering, Yale University, New Haven, Conn.

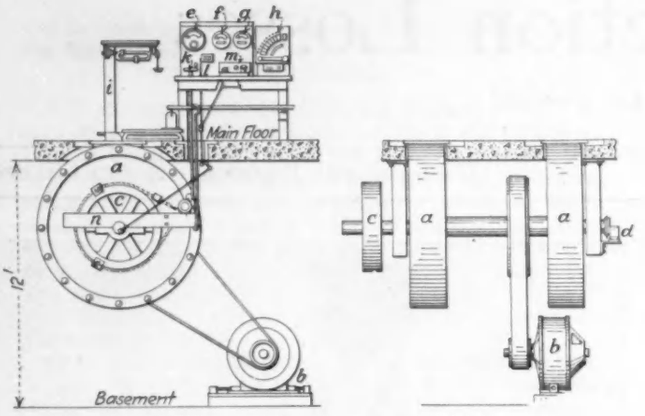

FIG. 3—VERTICAL SECTION OF THE APPARATUS

i Platform scales for weighing the pull on the brake-arm
k Control-handle for adjusting the brake-band tension

MEASUREMENT OF ROLLING FRICTION

The simplest measure of rolling friction is the tractive force that must be applied to the axle of the wheel to overcome the resistance. The tractive force can be measured by a spring-balance when the wheel is being towed, as shown in Fig. 4 at the left. The towing method is not practicable except at very slow speeds. The drum method, shown in the right-hand portion of Fig. 4, produces the same result as the towing method, but avoids its drawbacks. In this case the wheel rotates with the axis stationary, while the tractive force required to revolve the drum is being measured. The tractive force at the drum circumference can be measured conveniently and accurately by the dynamometers on the Mason Laboratory drums. The rolling friction, or rolling resistance, is measured in pounds, and is the tractive force required to revolve the drums against the resistance of the wheels. The measurement of rolling resistance requires two separate steps: (*a*) the drums are revolved with the wheels in place, while the electric input is carefully read and recorded; during this stage, the current input represents the total power required to drive the drums and the vehicle wheels; (*b*) the wheels are removed from the drums, after which the idle drums are rotated at the same speed as before. The operator then adds the load to the prony brake until he has duplicated the original readings of the electric meters. When this has been done, it is plain that the *added load* in the second step is equal to the *subtracted load* in the first step, due to the removal of the wheels. In other words, the added load on the prony brake, expressed in pounds at the drum circumference, is a measure of the rolling resistance of the wheels.

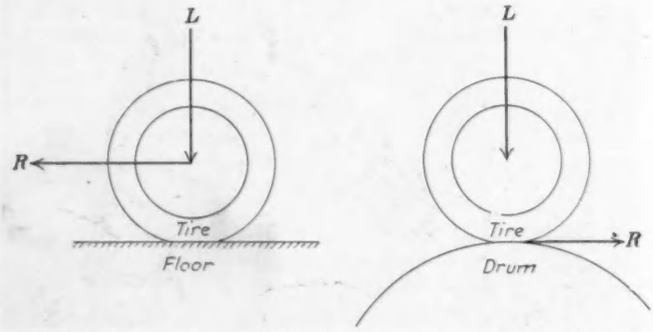

FIG. 4—DIAGRAM SHOWING THE TOWING AND THE DRUM METHODS OF MEASURING ROLLING FRICTION

This method can be varied to give the tire resistance alone, without the resistance of the bearings. To do this the wheels are not removed from the drums, but are jacked up until the rubber surface rests lightly upon the drums with contact enough to cause the wheels to revolve. The second step is then repeated, load being added to the prony brake until the original readings of the current meters are duplicated. The increase in the prony-brake load in this case will be found to be smaller than before, since it represents not the total resistance of the wheels, but only that portion of it that is due to the flattening of the tires on the drums. In jacking up the wheels for tire-resistance measurements, no definite pressure between the tire and the drum is required. Tests prove that any light pressure will suffice, provided it does not cause appreciable flattening of the rubber, yet drives the wheel.

This method was adopted because of its simplicity. It has proved sensitive in practice and readily allows small variations to be detected when the inflation-pressure or the load is changed. The method, however, involves a slight error for which no allowance has been made. This lies in the assumption that the drum-bearing friction and the wheel-bearing friction remain unchanged when the wheel is jacked up for the second reading. It is evident that the bearing friction must be less when the wheels are jacked up, hence the prony-brake reading represents not only the tire resistance but also the slight change in the bearing resistance. The final result, therefore, is slightly too large. It has been deemed safe to neglect this error since it is very small and does not affect the validity of comparative conclusions.

TOTAL FRICTIONAL RESISTANCE

Friction measurements have been made at intervals since 1916 on cars having great variations of weight and tire equipment. About 50 typical examples have been chosen for the comparison of the ratio of the total friction loss to the weight of the vehicle. These cars had pneumatic tires and ranged in weight from 1800 to 5300 lb. Fabric tires were used on the lighter cars, in most cases, and cord tires on the heavier ones. The inflation-pressure was from 60 to 80 lb. per sq. in. Observations were made at speeds of 20, 30 and 40 m.p.h. The results were averaged because the friction was practically constant at all speeds.

The front-wheel and the rear-wheel resistances were measured separately, and the results were plotted with the weight as the base line. Variations were found in cars of the same weight, as might have been expected. These are shown in Fig. 5 by the dots surrounding the line marked "front." An average line was plotted by taking the mean of the results for a group of cars having similar weights. These points are indicated by the encircled dots, the number showing the size of the group. A straight line was found to represent the tractive frictional resistance of both the front and the rear wheels of the car, after which they were combined into a single line marked "total."

The total frictional resistance of the average car as shown by these tests can be represented by the simple formula

$$R = 30 + 0.012\,L \qquad (1)$$

where

R = Total frictional tractive force, in pounds
L = Total weight of the vehicle, in pounds

The results given by the formula and the diagram in

Fig. 5 refer to the total frictional resistance of the tires, bearings and transmission on a smooth road. The resistance due to the wind is not included. The results given by the formula are for the average car and a considerable variation from this average can be expected. An easy-rolling car with cord tires may have, perhaps, 15 lb. less resistance than that given by the formula, while a hard-running car may have more.

The frictional tractive force of vehicles frequently has been expressed in the special unit "pounds per ton." This unit is convenient when the tractive force varies in direct proportion to the weight of the vehicle; otherwise it is not convenient. For example, in the group of 50 cars referred to, the "pounds per ton" varied from 54 to 34. This unit is somewhat awkward and open to ambiguity since the ton has two recognized values, 2000 lb. and 2240 lb., one being used mostly in America and the other in England.

It seems proper to suggest a new unit that is not ambiguous, namely, "pounds per thousand pounds," or "pounds per M." This unit is merely one-half the pounds per ton when the net ton is used. For example, 40 lb. per ton is the same as 20 lb. per M. Moreover, the new unit is directly comparable with the usual coefficient of friction by changing the decimal point. Thus 20 lb. per 1000 lb. becomes 20 lb. per M; and the frictional coefficient is therefore 0.020.

Resistance of Trucks

The frictional resistance of a number of heavy trucks equipped with both solid-rubber and pneumatic tires has been measured. These results, shown in Table 1, are interesting for comparison with passenger cars. The trucks have additional interest, as they belong to a group that is now being used for the measurement of tractive resistance under actual road conditions. The road tests are being carried out under the direction of Major Mark L. Ireland, of the Quartermasters' Corps, as a part of the extensive program of the advisory board of highway research of the National Research Council. It is expected that the tractive resistance of these trucks, as determined on the road, will soon be available for comparison with the laboratory dynamometer tests of the same trucks:

The trucks have been chosen as examples of the widest variation in internal friction. Yet the total tractive resistance, expressed in pounds per M, lies between the limits of 19.0 and 26.0. The higher values are clearly the fault of excessive friction in the transmission system, due probably to newness of the vehicle; hence they are likely to decrease with use.

These tests tend to prove that the rolling resistance of heavy trucks in good working order might be expressed as 20 to 21 lb. per M for all sizes. This figure differs from that of the group of 50 cars referred to in the previous paragraph and is expressed in equation (1). It agrees better with the group of seven cars mentioned in the next paragraph. These cars were tested recently and were known to be in free-rolling condition. Their rolling resistance was expressed by the figure, 19 lb. per M. This fairly close agreement among widely differing vehicles may lead to the further generalization that the rolling resistance of all rubber-tired vehicles, pneumatic and solid, when in best running condition, lies between the narrow limits of 19 and 21 lb. per M.

The lack of agreement with the 50 cars tested from 1916 to 1921 has been noted. This may be explained partly by the fact that most of the lighter cars were equipped with fabric tires, while the heavy cars were

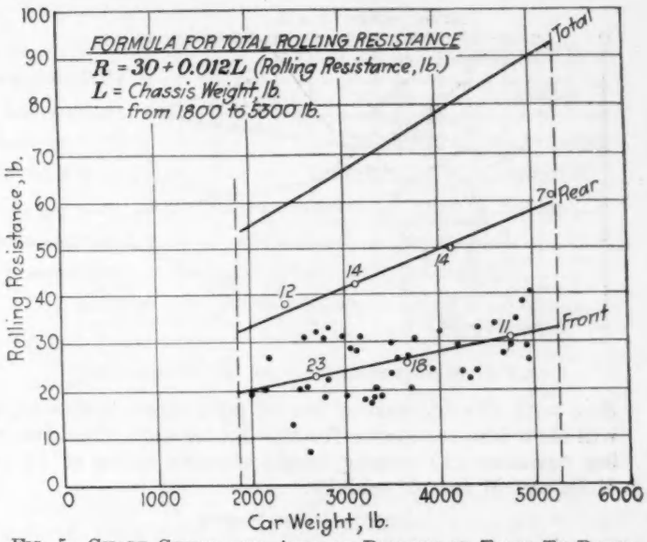

Fig. 5—Chart Giving the Average Results of Tests To Determine the Rolling Resistance of the Chassis of 50 Cars Equipped with Pneumatic Tires

equipped with cord tires. Equation (1) yields 27 lb. per M for light cars but, on the other hand, it gives 19 lb. per M for a car weighing 4500 lb. It seems likely

Table 1—Frictional Resistance of Trucks

Specification	Quartermaster Standard Type B Truck		Mack Chain-Drive 7½-Ton Truck
	No. 432,799	No. 44,913	
Front Wheels			
Weight, lb.	4,665	4,395	5,200
Tires, Solid Rubber Single, in	36x5	.....	36x7
Tires, Pneumatic Single, in.	.....	38x7	.....
Total Tractive Force, lb.	80.0	63.0	98.0
Tractive Force per 1,000 Lb. of Weight, lb.	17.5	14.4	19.0
Rear Wheels			
Weight, lb.	6,875	6,300	7,115
Tires, Solid Rubber Dual, in.	40x6	.....	40x7
Tires, Pneumatic Single, in.	.....	44x10	.....
Total Tractive Force, lb.	155.0	215.0	195.0
Tractive Force per 1,000 Lb. of Weight, lb.	22.5	34.0	27.5

Loaded Truck, Complete

	Weight, Lb.	Total Tractive Force, Lb.	Tractive Force per 1,000 Lb. of Weight, Lb.
Type B Truck, No. 432,799	11,540	235	20.3
	14,540	292	20.0
	17,540	344	19.6
	20,540	396	19.3
Type B Truck, No. 44,913	10,695	278	26.0
	13,695	349	25.5
Mack 7½-Ton Truck	12,315	293	23.8
	15,315	340	22.2

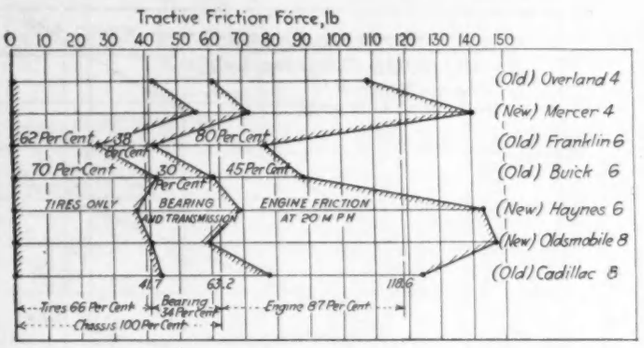

FIG. 6—INTERNAL FRICTION OF AUTOMOBILES

that with the increasing use of cord tires future tests will show less resistance for the lighter cars, thus bringing equation (1) nearer to the simpler value of 19 to 21 lb. per M for all vehicles.

FRICTION OF PARTS

Results have been given for the total chassis friction and its relation to the weight of the vehicle. It is now proposed to subdivide the total friction into several components to observe their relative importance better.

Five friction items have been chosen as giving the most useful information: (*a*) the front tires, (*b*) the front bearings, (*c*) the rear tires, (*d*) the rear bearings, including the transmission in the neutral gear; and (*e*) the engine friction in the direct drive when the rear wheels are turning at the equivalent of a car speed of 20 m.p.h.

Speed has not been mentioned in the first four items inasmuch as the friction is nearly constant at all speeds. Engine friction has been measured by driving through the rear wheels from the dynamometer, with the ignition shut-off. Care was taken to have the engine warm before beginning the measurements; also to have the throttle wide-open, as this was found to reduce the resistance materially. This last fact suggests that engine resistance is due in part to compression action in the cylinders or to air friction in the valves. It was observed that the engine resistance, measured in terms of tractive force at the rear wheels, increased directly as the speed. The frictional resistance of the engine at 40 m.p.h. was *double* that at 20 m.p.h. This result is significant by contrast with other friction measurements including those of tires, bearings and transmission, of which the increase of friction with speed is negligible. Granting that the engine friction determined in this way is too large, it may at least be used for comparative purposes.

Seven representative automobiles were chosen for the friction measurements. Table 2 gives the dimensions, weight, and the like, of the several cars, together with the friction items already mentioned. Considerable variation was found in the different cars, as might have been expected from the difference in weight. To bring this out more clearly, the total friction, divided into the three parts of tires, bearings and transmission, and the engine, has been plotted in Fig. 6.

A study of the friction diagram shows that, in spite of variations of other factors, the ratio of tire friction to bearing-transmission friction remains constant in the order of 2 to 1. In other words, the friction loss in the tires is two-thirds the total chassis friction, exclusive of the engine.

Examination of the internal-friction chart reproduced in Fig. 6 shows that the item of engine friction is large, being on the average seven-eighths the remaining chassis friction. Admitting that the measured engine frictions are too large, for reasons already mentioned, it is probably true that comparisons can be made fairly between them. Three of the engines had been used but little since leaving the factory and showed higher friction than the others. Two had run over 12,000 miles since the last overhauling and were in free-running condition. Comparing the two groups, the new engines had nearly three times the friction of the old ones.

A true comparison of the frictional resistance of the chassis requires that all values be reduced to the same car-weight. Table 2 is made out for the seven cars, giving the resistance of the tires, bearings and transmission, and chassis without the engine, in terms of pounds per thousand pounds. The items for the different cars are by no means constant, yet they are not widely apart. The average of the three columns can be considered as fairly representative of the friction items for the cars weighing from 3000 to 5000 lb. From these results the following conclusions can be drawn: In cars weighing from 2500 to 5000 lb. the average friction of cord tires only was 12.5 lb. per M; of bearings and transmission only, 6.5 lb. per M; and of total chassis without the engine, 19 lb. per M.

TABLE 2—FRICTIONAL DISTRIBUTION IN THE CHASSIS

Date of Test	April 3, 1922	April 4, 1922	April 10, 1922	April 11, 1922	April 11, 1922	April 12, 1922	April 12, 1922
Name of Car	Overland	Mercer	Buick	Cadillac	Oldsmobile	Franklin	Haynes
Car Model	85	.....	44	.....	47 FS	9A	55
Number of Cylinders	4	4	6	8	8	6	6
Engine Bore and Stroke, in.	4 1/8x4 1/2	3 3/4x6 3/4	3 1/4x4 1/2	3 1/8x5 1/8	2 7/8x4 1/2	3 1/4x4	3 1/2x5
Weight: Front Wheels, lb.	1,360	1,850	1,350	2,010	1,495	1,250	1,585
Rear Wheels, lb.	1,650	2,250	1,460	2,400	1,735	1,450	1,815
Total, lb.	3,010	4,100	2,810	4,410	3,230	2,700	3,400
Front Tires: Size, in.	32x4	32x4 1/2	32x4	35x5	32x4	32x4	33x4
Kind	1 Fabric 1 Cord	Goodyear Cord	1 Fabric 1 Cord	Revere Cord	Federal Cord	Goodyear Cord	Goodyear Cord
Rear Tires: Size, in.	32x4	32x4 1/2	32x4	35x5	32x4	32x4	33x4
Kind	Cord	K-S Cord	1 Fabric 1 Cord	Yale Cord	Federal Cord	Goodyear Cord	Goodyear Cord
Rolling-Resistance:							
Front Tires, lb.	23.2	25.3	22.8	21.0	16.4	9.5	14.6
Front Bearings, lb.	4.4	2.4	3.2	6.2	4.0	7.4	3.9
Total Front Wheel, lb.	27.6	27.7	26.0	27.2	20.4	16.9	18.5
Rear Tires, lb.	19.0	30.0	20.3	23.5	25.2	16.0	22.8
Rear Bearings and Transmission, lb.	14.0	14.0	14.6	26.5	13.8	10.0	27.8
Total Rear Wheel, lb.	33.0	44.0	34.9	50.0	39.0	26.0	50.6
Total, Front and Rear Wheels, lb.	60.6	71.7	60.9	77.2	59.4	42.9	69.1
Tires Only, lb	42.2	55.3	43.1	44.5	41.6	25.5	37.4
Bearings and Transmission, lb.	18.4	16.4	17.8	32.7	17.8	17.4	31.7
Engine, at 20 M.P.H., lb.	47.5	69.0	27.4	47.5	88.0	34.0	74.4
Total, Including Engine, lb.	108.1	140.7	88.3	124.7	147.4	76.9	143.5

Resistance Formulas

Two rules for determining the total chassis friction have been deduced from the dynamometer tests. Their agreement logically should be compared. The first was deduced from 50 tests of various cars where the front and the rear resistances were measured separately. This embraced a variety of tire equipment that was mostly fabric tires on the lighter cars and cord tires on the heavier ones. These tests were spread over a period of 5 years. The formula for the first series, where L = the weight of the chassis, is

$$\text{Total chassis resistance} = 0.012\,L + 30$$

The second series consisted of seven cars, where the tire and the bearing resistances were separately measured. Practically all the cars were equipped with cord tires and the measurements were made at one time by the same observers. The formula for the second series is

$$\text{Total chassis resistance} = 0.019\,L$$

TABLE 3—INTERNAL FRICTION AND TOTAL FRICTION, IN POUNDS OF TRACTIVE FORCE PER 1000 LB. OF WEIGHT

Name of Car	Weight, lb.	Internal Friction, Lb.		Total Friction, Lb.
		Tires	Bearings and Transmission	
Buick	2,810	15.4	6.3	21.7
Cadillac	4,410	10.1	7.5	17.6
Franklin	2,700	9.5	6.4	15.9
Haynes	3,400	11.0	9.3	20.3
Mercer	4,100	13.6	4.0	17.6
Oldsmobile	3,230	13.0	5.4	18.4
Overland	3,010	14.0	6.0	20.0
Average	3,380	12.4	6.4	18.8

The divergence of the two formulas can be seen best by comparing the values for the same weights of car. For L = 2500, the resistance by the two formulas is: first, 60 lb.; second, 47.5 lb.; difference, 12.5 lb. For L = 4000, the two formulas give first, 78 lb.; second, 76 lb.; difference, 2 lb. From this comparison it appears that the first formula gives larger values for the light-weight cars, while both agree well for the heavier cars. The theory has been advanced that the general use of fabric tires on the lighter cars was the cause of their increased resistance.

The internal friction of several heavy trucks has been separated into tire and bearing elements as shown in Table 4.

In Table 4, Truck B 432,788 was known to have the excessive transmission friction incident to a new truck, which is shown by its bearing and transmission friction of 12.0 lb. per M or nearly double that of its companion truck. The other two trucks can be considered as representatives of the heavy class. Their average tire friction is 13.5 lb. per M, or 61 per cent; average bearing and transmission friction, 8.5 lb. per M, or 39 per cent; total, 22.0 lb. per M, or 100 per cent.

The corresponding figures for passenger cars with cord tires were: average tire friction, 12.5 lb. per M, or 66 per cent; average bearing and transmission friction, 6.5 lb. per M, or 34 per cent; total, 19.0 lb. per M, or 100 per cent. In general, the friction distribution in trucks does not differ greatly from that in lighter vehicles.

TABLE 4—INTERNAL FRICTION OF HEAVY TRUCKS

Name of Truck	Tractive Resistance in Percentage and Pounds per 1000 Lb. of Weight		
	Tires Only	Bearings and Transmission	Total
Quartermaster Standard, Type B, No. 432,799	13.7 lb. 67%	6.6 lb. 33%	20.3 lb. 100%
Quartermaster Standard, Type B, No. 432,788	15.1 lb. 56%	12.0 lb. 44%	27.1 lb. 100%
Mack 7½-Ton, Type-AC, Chain Drive	13.4 lb. 56%	10.4 lb. 44%	23.8 lb. 100%

Fabric Tires

The influence of load and inflation-pressure on the rolling resistance of pneumatic tires becomes evident when the results of tests are studied. Some typical examples will now be presented for a line of fabric tires from the same maker, sizes 32 x 4 in., 33 x 4½ in. and 35 x 5 in. In making tests of the rolling resistance of these tires, an arbitrary schedule of loads and inflation-pressures was adopted, designed to cover a sufficiently wide range for the size of the tires. Three different speeds were used for each test-load and inflation-pressure, resulting in as many as 36 independent readings of rolling resistance for one tire. The changes due to speed were very slight; hence it was possible to eliminate the speed as a variable, giving an average value for each load and inflation-pressure without regard to the speed. The rolling resistance of each tire, arranged in columns under the respective loads, is given in Table 5. The variation of the figures in each column is a measure of the effect produced by the change of inflation pressure.

To bring out the variation more clearly, a parallel column has been added to Table 5 giving the results in

TABLE 5—ROLLING-RESISTANCE OF FABRIC NON-SKID TIRES AT 20 TO 40 M.P.H. SPEEDS

Test No. 28; Size, 32x4 In.; Weight of Tire, 19.5 Lb.; Outside Diameter 32.8 In.

Inflation-Pressure, Lb. per Sq. In.	Load, 460 Lb.		Load, 700 Lb.		Load, 975 Lb.	
	Rolling-Resistance		Rolling-Resistance		Rolling-Resistance	
	Lb.	Per Cent	Lb.	Per Cent	Lb.	Per Cent
30	8.03	186	14.45	165	22.3	163
55	5.35	124	10.60	121	16.8	121
80	4.33	100	8.78	100	13.7	100

Test No. 60; Size, 33x4½ In.; Weight of Tire, 23.5 Lb.; Outside Diameter, 33.7 In.

	Load, 585 Lb.		Load, 935 Lb.		Load, 1,285 Lb.	
30	9.05	124	18.40	152	27.7	145
45	8.65	118	15.60	130	23.0	121
65	8.65	118	14.20	118	21.2	111
90	7.32	100	12.03	100	19.1	100

Test No. 102; Size, 35x5 In.; Weight of Tire, 33.5 Lb.; Outside Diameter, 36.0 In.

	Load, 750 Lb.		Load, 1,050 Lb.		Load, 1,650 Lb.	
40	12.70	123	19.80	128	37.0	148
55	11.40	109	17.60	115	32.4	129
70	10.50	102	15.90	103	28.2	113
90	10.30	100	15.40	100	25.1	100

TABLE 6—COMPARATIVE ROLLING FRICTION OF FABRIC AND CORD TIRES

Size of Tire, in.	32x4	33x4½	35x5
Speed, m.p.h.	20 to 40	20 to 40	20 to 40
Inflation-Pressure, lb. per sq. in.	30 to 80	30 to 90	40 to 90
Load, lb.	460 to 975	585 to 1,285	750 to 1,650
Number of Readings	54	72	32
Average of Rolling-Friction Readings...			
Fabric, lb.	11.30	15.20	21.10
Cord, lb.	7.60	9.06	13.45
Fabric, per cent.	100.00	100.00	100.00
Cord, per cent.	67.00	60.00	64.00

percentages, using 100 per cent for the smallest value of the resistance. The increase of rolling resistance due to lower inflation-pressure is apparent in every case. A general conclusion from the figures in Table 5 can be stated thus: The rolling resistance of a fabric tire, fully loaded and inflated, may be increased by more than 50 per cent when the inflation-pressure is dropped from 90 to 30 lb. per sq. in., and by 25 per cent when the pressure is dropped to 50 lb. per sq. in. A word of caution is added in this connection, namely, that the total car resistance will not be increased in this same ratio, because

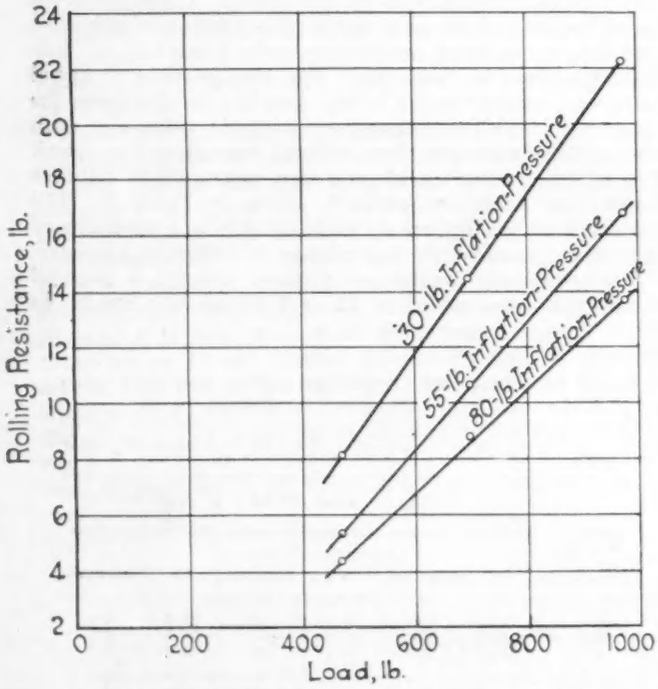

FIG. 7—AVERAGE ROLLING RESISTANCE OF A 32 X 4-IN. FABRIC PNEUMATIC TIRE AT SPEEDS OF FROM 20 TO 40 M.P.H. AND DIFFERENT INFLATION-PRESSURES

the tires produce only a part of the total friction of the chassis; also, that this result applies only to *fabric* tires.

The diagrams shown in Figs. 7, 8 and 9, contain a curve for each inflation-pressure, plotted with the rolling resistance and the load as coordinates. The plotted points show the existence of slight observational errors, yet fairly satisfactory curves can be drawn through them. An interesting fact, shown on all the diagrams, is that the line for all the inflation-pressure curves is straight. Another is that, at the same inflation-pressure, these straight lines practically coincide on each diagram, showing that the rolling resistance depends solely upon the load and not upon the size of the tire. Whether this conclusion can be extended to the larger sizes, such as pneumatic tires for trucks, will be discussed in another paragraph.

The rolling resistance of the fabric tires, when fully

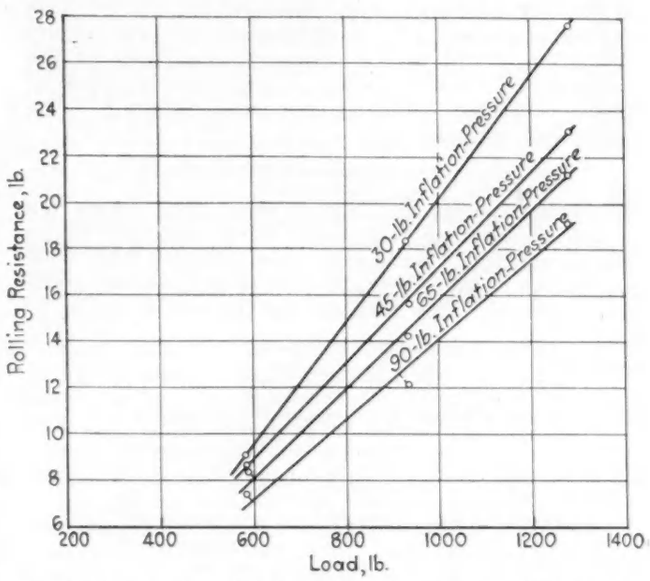

FIG. 8—AVERAGE ROLLING RESISTANCE OF A 33 X 4½-IN. FABRIC PNEUMATIC TIRE AT SPEEDS OF FROM 20 TO 40 M.P.H. AND DIFFERENT INFLATION-PRESSURES

inflated, can be expressed by a simple algebraic formula which applies to all three sizes:

$$R = 0.018\,L - 3.0 \qquad (2)$$

where,

R = the rolling resistance, in pounds
L = the load on the tire, in pounds

Another way of expressing the result is in the simple form of pounds per thousand pounds. This number

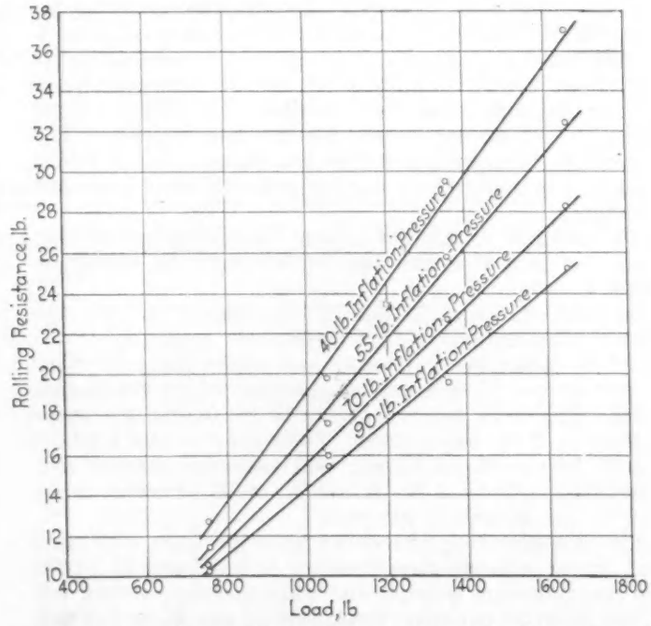

FIG. 9—AVERAGE ROLLING RESISTANCE OF A 35 X 5-IN. FABRIC PNEUMATIC TIRE AT A SPEED OF 20 M.P.H. AND DIFFERENT INFLATION-PRESSURES

varies slightly with the load. Its average value is 15 lb. per M for the tire sizes referred to in this article.

Cord Tires

Dynamometer tests of cord tires show less rolling resistance than those of fabric tires. The difference can be stated as being approximately one-third. In other words, the rolling resistance of a standard make of cord tire is only two-thirds that of a fabric tire. This conclusion has been proved many times during the past 5 years with different sizes and makes of tire. The figures in Table 6, based on tests of Fisk and Goodyear tires, both of which have shown practically identical rolling resistances, are presented as typical.

The deductions from these figures are confirmed by curves plotted from the test results. An example is given in Fig. 10, where two rolling-resistance curves are

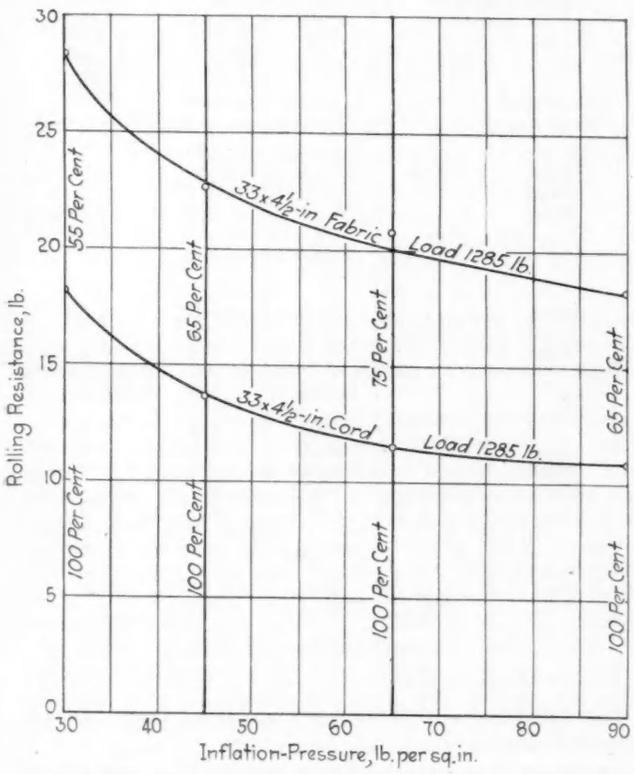

Fig. 10—Typical Rolling-Resistance Curves of Fabric and Cord Tires at a Constant Load and Varying Inflation-Pressures

plotted, one for fabric and one for cord tires. The curves are nearly parallel, showing that the difference in friction is about the same at all inflation-pressures.

The cord tire has a valuable characteristic that is clearly shown in Fig. 10. A low friction is reached at a moderate inflation-pressure. In this case the resistance is 11.5 lb. at an inflation-pressure of 65 lb. per sq. in., and 11.0 lb. at a pressure of 90 lb. per sq. in.; hence, the cord tire can be run at a moderate inflation-pressure with only a slight loss of power. On the other hand, the fabric-tire resistance diminishes steadily as the inflation-pressure is increased; hence, high pressures are demanded for the saving of power.

Another illustration of the difference between fabric and cord-tire resistances is given in Fig. 11. In this example the curves of rolling resistance are plotted for a constant inflation-pressure. The results are given for several sizes of fabric and cord tires. There are differ-

Table 7—Average Rolling-Friction for Seven Ribbed and Non-Skid Firestone, Fisk, Goodrich and Goodyear Tires

Inflation-Pressure, Lb. per Sq. In.	Load, Lb.	Speed, M.P.H. 20	30	40	Average Rolling-Friction, Lb.
30	585	6.03	6.33	6.91	6.42
45		5.16	4.98	5.26	5.13
65		4.68	4.70	4.65	4.68
90		4.17	4.29	4.54	4.33
Average					5.14
30	935	10.99	11.75	12.37	11.70
45		9.04	9.76	10.01	9.60
65		8.21	8.34	8.43	8.33
90		7.49	8.01	8.22	7.90
Average					9.38
30	1,285	17.39	18.17	19.01	18.19
45		14.06	14.71	14.89	14.55
65		12.16	12.32	12.64	12.37
90		10.87	11.30	11.70	11.29
Average					14.10
General Average					9.54

ences between the various sizes, but the points lie in a band, or strip, in each case. These curves show clearly that the rolling resistance of any size of cord tire, under any load, is about two-thirds that of a fabric tire under the same conditions.

The rolling resistance of cord tires is further analyzed in Table 7, which gives the rolling friction of 33 x 4-in. tires arranged for three speeds, three loads and four inflation-pressures, a total of 36 readings. The figures in Table 7 are compiled from tests of seven cord tires of well-known makes, and therefore can be considered as fairly representative of new cord tires.

The tabular values have been plotted in the diagram, Fig. 12, with one curve for each 10 lb. of inflation-pressure. The spacing of the curves shows clearly the change of resistance with any change of pressure. After reaching an inflation-pressure of 60 lb. per sq. in., but

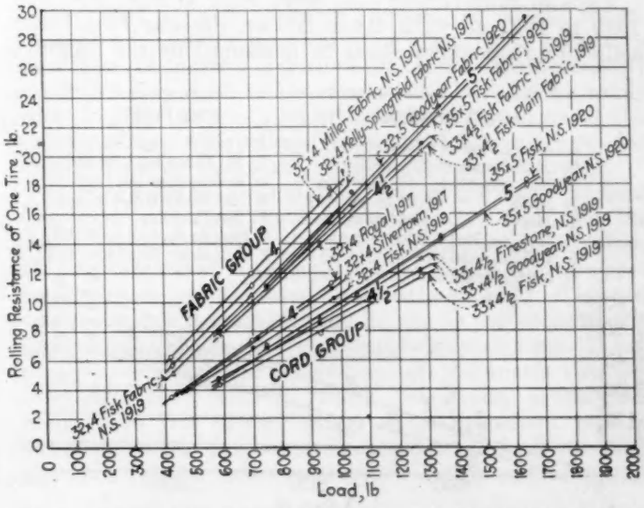
Rolling Resistance of One Tire, lb.
Load, lb
FABRIC GROUP
CORD GROUP
32 x 4 Fisk Fabric, N.S. 1919
32 x 4 Miller Fabric, N.S. 1917
32 x 4 Kelly-Springfield Fabric N.S. 1917
32 x 4 Royal, 1917
32 x 4 Silvertown, 1917
32 x 4 Fisk, N.S. 1919
33 x 4½ Fisk Fabric, 1920
33 x 4½ Fisk Fabric, N.S. 1919
33 x 4½ Fisk Plain Fabric, 1919
35 x 5 Goodyear Fabric, 1920
35 x 5 Fisk Fabric, N.S. 1919
35 x 5 Fisk, N.S. 1920
35 x 5 Goodyear, N.S. 1920
33 x 4½ Firestone, N.S. 1919
33 x 4½ Goodyear, N.S. 1919
33 x 4½ Fisk, N.S. 1919

Fig. 11—Comparative Rolling-Resistance Curves of Fabric and Cord Tires at an Inflation-Pressure of 65 Lb.

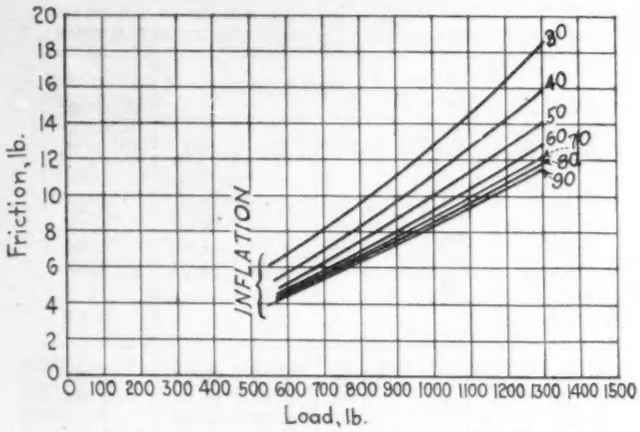

FIG. 12—ROLLING-FRICTION OF 33 X 4½-IN. CORD TIRES AT DIFFERENT LOADS AND INFLATION-PRESSURES RANGING FROM 30 TO 80 LB.

little further decrease in rolling friction is found, again showing that in cord tires a high degree of inflation is not required for easy rolling.

The resistance curves are practically straight lines of varying slope. An empirical formula has been derived for these curves that gives the rolling resistance for any load and inflation-pressure within the limits covered by the experimental work

$$R = [(L - 200) \div 100] \times \{ [1.06 + (90 - P)^2] \div 7000 \} \quad (3)$$

where

R = Rolling-resistance
L = Load
P = Inflation-pressure

The limits of load are from 500 to 1300 lb., and those of inflation pressure, 30 to 90 lb. per sq. in. This formula can be put into simpler form by inserting a normal inflation-pressure of 60 lb. per sq. in., causing it to take the form

$$R = 0.012\,L - 2.0 \quad (4)$$

A similar formula was deduced for fabric tires, where the constants are exactly 50 per cent larger than for the cord-tire formula. This is further evidence that cord tires have two-thirds the rolling friction of fabric tires.

SOLID-RUBBER TIRES

Tests of solid-rubber tires were made at intervals from 1919 to 1921 on ¾ to 2-ton trucks. It was found that solid-tire resistances could be measured by the jacking-up method and that, in general, the resistance was between those of cord and fabric tires under the same load.

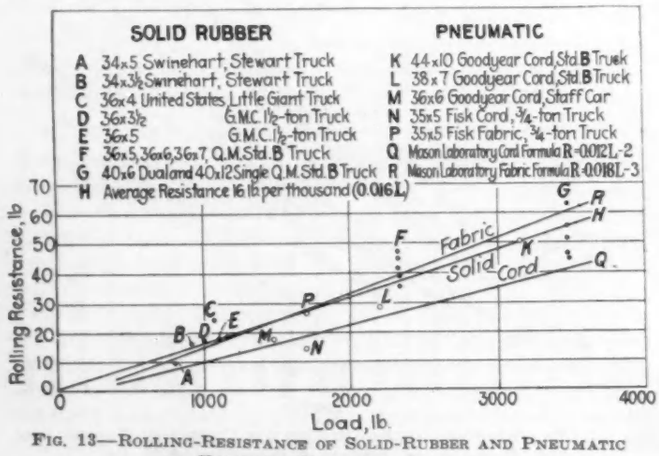
SOLID RUBBER
A 34x5 Swinehart, Stewart Truck
B 34x3½ Swinehart, Stewart Truck
C 36x4 United States, Little Giant Truck
D 36x3½ G.M.C. 1½-ton Truck
E 36x5 G.M.C. 1½-ton Truck
F 36x5, 36x6, 36x7, Q.M. Std. B Truck
G 40x6 Dual and 40x12 Single Q.M. Std. B Truck
H Average Resistance 16 lb per thousand (0.016 L)
PNEUMATIC
K 44x10 Goodyear Cord, Std. B Truck
L 38x7 Goodyear Cord, Std. B Truck
M 36x6 Goodyear Cord, Staff Car
N 35x5 Fisk Cord, ¾-ton Truck
P 35x5 Fisk Fabric, ¾-ton Truck
Q Mason Laboratory Cord Formula R=0.012L-2
R Mason Laboratory Fabric Formula R=0.018L-3
Rolling Resistance, lb
Load, lb.
Fabric
Solid
Cord

FIG. 13—ROLLING-RESISTANCE OF SOLID-RUBBER AND PNEUMATIC TIRES FOR DIFFERENT LOADS

In 1921, in cooperation with the Committee on the Tractive Resistance of Roads Research, the drums were widened to accommodate the dual tires of heavy trucks. Six trucks were brought to the Mason Laboratory for the investigation of internal friction by the dynamometer method. As planned by Major Ireland, the tests were carried out with empty and with loaded trucks, with different temperatures of the lubricant, with several kinds of new solid-rubber tires, and with a few well-worn tires. The tire-resistance measurements are summarized as shown in Table 8.

TABLE 8—SUMMARY OF TIRE-RESISTANCE MEASUREMENTS

		Tire Size, In.	Load per Inch of Tire Width, Lb.	Tire Resistance per 1,000 Lb. of Weight, Lb.
New Front Tires	Single	36x4	583	21.7
		36x5	466	19.0
		36x6	375	20.2
		36x7	375	19.0
		Average..	450	20.0
Worn Front Tires	Single	36x5	466	15.5
		36x6	375	16.5
		36x7	375	16.2
		Average..	400	16.1
New Rear Tires	Dual	40x[illegible]	2[illegible]5	18.0
		40x6	275	15.7
	Single	40x12	275	14.0
		40x12	275	15.8
		40x12	275	19.0
		Average..	275	16.5
Worn Rear Tires	Dual	40x6	285	12.5
		40x6	255	11.0
		Average..	270	11.8

The following conclusions are apparent from the figures in Table 8:

(1) Partly worn tires roll with less friction than new and more resilient tires, the difference being from 20 to 25 per cent; hence, increased cushioning must be paid for by increased power

(2) Front tires show more friction than rear tires, both when new and when worn. This is to be explained by the relatively heavier load carried by the front tires. The rear-tire tests were made with the truck empty; hence, the load per inch of tire width was considerably less than that for the front wheels

(3) The rolling resistance of the solid-rubber tires varies considerably, depending upon the wear, the construction and the like, but the values are all between the cord and the fabric-tire figures. Taking the average of all the readings as an index, the solid-tire resistance is 16.0 lb. per M

A final comparison of solid and pneumatic tires has been made in Fig. 13. All the solid-rubber tire resistances have been plotted with respect to the load in three groups; for light trucks, and for the front and rear wheels of heavy trucks. The average of all the solid-

rubber-tire results has been represented by the line marked 16 lb. per M, as mentioned above. The results for several large-sized pneumatic-tires also are plotted in Fig. 13 and the Mason Laboratory formulas for fabric and cord tires are stated. Fig. 13 confirms the previous statement that solid-rubber-tire friction is approximately equal to fabric-tire friction.

Dynamometer tire-friction measurements have been made at speeds of 20, 30 and 40 m.p.h. Over this range the observed readings have been nearly constant. Stated more correctly, the variations due to speed have been of about the same order as the observational error of a single reading; hence they are not easily observed. When the averages of many different readings are taken,

TABLE 9—AVERAGE ROLLING-RESISTANCE AT DIFFERENT SPEEDS FOR 11 33x4½-IN. TIRES

Speed, M.P.H.	Series No. 1		Series No. 2	
	Inflation-Pressure, 65 Lb. per Sq. In.		Inflation-Pressure, 90 Lb. per Sq. In.	
	Load, 935 Lb.		Load, 1,285 Lb.	
	Average Resistance		Average Resistance	
	Lb.	Per Cent	Lb.	Per Cent
20	9.4	100	12.6	100
30	9.8	104	13.1	104
40	9.9	106	13.6	108

a slight increase of the rolling resistance with the speed is found. The figures in Table 9 show that, by doubling the speed, the rolling resistance may be increased by 6 or 8 per cent.

In view of the small change produced by the speed, it has been deemed most satisfactory to ignore it and to give the average value as constant for all speeds. This assumption is allowable for rolling on smooth, hard surfaces. For rough, uneven or soft roads, the rolling resistance doubtless will increase materially with the speed.

Careful measurements have shown that the non-skid type of tread produces slightly more resistance than the smooth tread. The increase produced by the non-skid tread is small and is presented as a matter of interest rather than importance. The figures in Table 10, based on tests of three well-known tires, will indicate the influence of the non-skid tread.

The figures in Table 10 seem to show that the effect of the non-skid tread is to increase the rolling resistance about 5 per cent. The increase is caused, probably, by

TABLE 10—INFLUENCE OF NON-SKID TREAD ON ROLLING-RESISTANCE[2]

Kind of Tire, Cord	Rolling-Resistance	
	Ribbed Tread, Lb.	Non-Skid, Tread, Lb.
Fisk	8.86	9.23
Firestone	8.84	9.52
Goodyear	8.80	9.98
Average	8.83	9.58

[2] Average of 36 readings on 33 x 4½-in. cord tires at different speeds, loads and inflation-pressures. The increase in resistance due to the non-skid tread is 8 per cent.

TABLE 11—ROLLING-RESISTANCE OF LARGE PNEUMATIC TIRES

Tire Size, In.	Rolling-Resistance per 1,000 Lb. of Weight, Lb.
32x4	11.0
33x4½	11.0
35x5	11.0
36x6	12.5
38x7	13.0
44x10	16.0

the greater compression of the rubber under the load and may vary somewhat with the shape of the non-skid markings.

The influence of heavy-wall tubes of the "non-puncture" type has been measured by running alternate tests with thin-wall and thick-wall tubes in the same casing. The results in two instances have shown slightly greater friction for the thick-wall tubes, the difference being about 5 to 10 per cent.

Similar results have been obtained from a tube protector, consisting of a thick endless pad lying between the tube and the casing. The protecting strip was found to increase the rolling resistance by 7 per cent, in a 33 x 4½-in. cord-tire casing.

Large Pneumatic Tires

Test of the smaller sizes, 4 and 5-in. tires, have shown that the rolling resistance is practically identical for the same loads and inflation-pressure. If this holds true, it

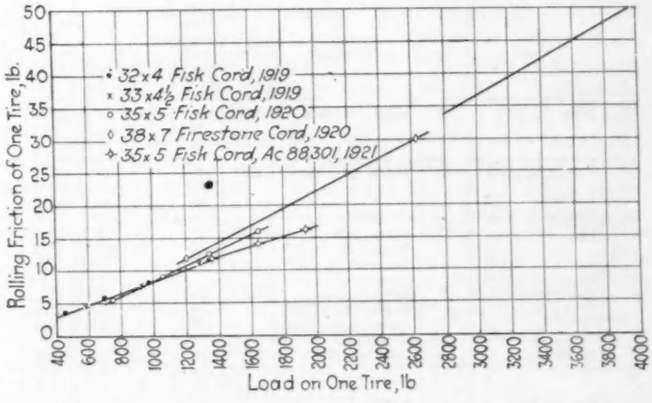
Rolling Friction of One Tire, lb.
Load on One Tire, lb
32x4 Fisk Cord, 1919
33x4½ Fisk Cord, 1919
35x5 Fisk Cord, 1920
38x7 Firestone Cord, 1920
35x5 Fisk Cord, Ac 88301, 1921

FIG. 14—ROLLING-RESISTANCE OF DIFFERENT SIZES OF CORD TIRES AT AN INFLATION-PRESSURE OF 90 LB.

follows that a single curve or formula for rolling resistance would hold for all sizes of a given type, such as a cord tire of normal inflation-pressure. Unfortunately for this theory, tests of the larger sizes have shown somewhat higher friction, proportionately, than those of the smaller sizes. The values in Table 11 are averages for several sizes of cord tires as measured on the drums.

Fig. 14 affords a comparison of several sizes of cord tire under different loads. It is probably true that the larger sizes were less well inflated than the smaller sizes. In one case, that of the 44 x 10-in. size, the tire was marked with the direction to "Inflate to 140." The test was made at a pressure of 100 lb. per sq. in. because of a limitation of the available air-supply at that time. It can be concluded that the larger-sizes of pneumatic tire will have somewhat higher rolling resistance, proportionately, than the smaller sizes. This statement can be qualified by the further assumption that if all the sizes were kept at the inflation-pressures recommended by the

manufacturers, the rolling resistances would be practically alike.

The evidence of the effect of wear on the rolling resistance of pneumatic tires is rather conflicting, since some old tires have shown excessive friction, while others have agreed well with that of new tires. Close inspection proves that in a majority of cases the rolling frictions of a new and of an old tire are about the same; also that in exceptional instances some reason for the difference usually can be found as when old tires get out of shape and run untrue on the rims. Reliable information on the effect of age and wear on tires is difficult to obtain but fortunately this is not a matter of practical importance. From the evidence at hand it is believed that a worn pneumatic tire has about the same rolling friction as a new tire of the same kind. Solid-rubber-tire resistance is known to decrease as the tire wears out, leaving a thinner layer of rubber.

The Mason Laboratory tests give evidence that there has been a general decrease in the rolling resistance of pneumatic tires for several years. Tests of tires of the same size have been carried on under the same conditions of loading and inflation; hence the results should be comparable. In the 32 x 4-in. size, a considerable decrease was observed from 1917 to 1919 in both fabric and cord tires as shown in Table 12. This has been partly lost between 1919 and 1922. The decrease in rolling friction from 1917 to 1919 is shown in detail in several of the diagrams of comparative results.

TABLE 12—ROLLING-RESISTANCE OF 32 X 4-IN. PNEUMATIC TIRES

Name	Number	Description	Year	Inflation-Pressure, Lb. per Sq. In	LOAD, LB. 500	700	1,000
Miller	3	Fabric, non-skid	1917	30	8.8	14.0	21.5
				55	7.2	11.7	18.5
				80	6.5	11.0	17.5
Kelly-Springfield	12	Fabric, non-skid	1917	30	9.1	15.5	25.0
				55	7.7	13.1	21.2
				80	6.3	10.5	17.1
Fisk	27	Fabric, plain Weight, lb. 18.0 Diameter, in. 32.6	1919	30	7.7	12.3	20.0
				55	6.5	10.5	16.5
				80	4.9	8.0	13.0
Fisk	28	Fabric, non-skid Weight, lb. 19.5 Diameter, in. 33.0	1919	30	8.8	14.4	22.8
				55	6.1	10.5	17.4
				80	5.0	8.6	14.0
Firestone	16	Cord, non-skid	1917	30	8.4	12.5	18.9
				55	6.1	9.3	14.0
				80	5.5	8.5	13.3
Goodrich	14	Cord, ribbed	1917	30	6.5	10.5	17.2
				55	5.5	8.0	12.5
				80	4.8	6.8	10.2
Goodyear	15	Cord, non-skid	1917	30	7.8	12.3	19.5
				55	6.0	8.9	13.6
				80	5.0	7.4	11.3
Miller	6	Cord, non-skid	1917	30	8.2	12.1	19.5
				55	5.2	8.0	14.0
				80	4.5	7.0	11.2
U. S. Royal	8	Cord, non-skid	1917	30	8.0	12.1	19.8
				55	4.9	7.5	12.7
				80	3.9	5.9	10.8
Fisk	26	Cord, ribbed Weight, lb. 21.0 Diameter, in. 33.3	1919	30	5.9	8.6	15.0
				55	4.5	6.5	10.6
				88	3.8	5.5	8.5
Fisk	25	Cord, non-skid Weight, lb. 21.5 Diameter, in. 33.3	1919	30	5.7	9.0	14.0
				55	4.7	7.2	11.4
				80	4.0	6.0	10.3
Fisk	29	Cord, non-skid Weight, lb. 22.0 Diameter, in. 33.3	1922	30	7.2	9.5	15.0
				55	6.3	8.0	11.7
				80	5.5	7.3	10.8
Fisk	30	Cord, non-skid Weight, lb. 22.2 Diameter, in. 33.3	1922	30	6.5	9.5	16.3
				55	5.0	7.6	12.5
				80	4.6	6.8	11.3
Yale	31	Cord, non-skid Weight, lb. 25.0 Diameter, in. 33.75	1922	30	9.0	12.5	22.0
				55	7.0	9.5	15.0
				80	6.5	8.5	13.0

TABLE 13—ROLLING-RESISTANCE OF 35 X 5-IN. PNEUMATIC TIRES

Name	Number	Description	Year	Inflation-Pressure, Lb per Sq. In	LOAD, LB. 700	1,000	1,300	1,600
Goodyear	100	Fabric, non-skid Weight, lb. 31.0 Diameter, in. 35.9	1920	40	12.8	20.0	27.5	35.0
				55	11.2	17.5	24.1	30.5
				70	10.0	15.8	21.7	27.7
				90	9.5	15.0	20.6	26.2
Fisk	102	Fabric, non-skid Weight, lb. 33.5 Diameter, in. 36.0	1920	40	11.5	19.6	27.8	36.0
				55	10.5	17.4	24.5	31.5
				70	9.7	15.6	21.5	27.3
				90	9.4	14.4	19.3	24.0
Yale	104	Fabric, non-skid Weight, lb. 36.25 Diameter, in. 36.0	1920	40	14.0	22.8	31.5	40.5
				55	12.7	19.5	26.2	33.0
				70	11.7	17.5	23.3	29.2
				90	10.7	15.5	20.4	25.2
Goodyear	101	Cord, non-skid	1920	40	9.0	13.5	18.5	23.2
				55	7.3	11.2	15.3	18.3
				70	6.0	9.6	13.5	17.3
				90	5.0	8.5	12.4	16.0
Fisk	103	Cord, non-skid Weight, lb. 35.5 Diameter, in. 37.2	1920	40	8.5	12.8	18.2	24.0
				55	6.8	11.1	15.5	20.0
				70	5.9	9.6	13.4	17.2
				90	4.8	8.5	12.0	15.5
Fisk	109	Cord, non-skid special Weight, lb. 37.0 Diameter, in. 37.2	1921	40	.....	11.5	15.5	20.0
				55	.....	10.3	13.6	17.0
				70	.....	9.0	12.0	14.7
				90	.....	8.3	11.0	13.5

The 35 x 5-in. size referred to in Table 13 was first tested in the Mason Laboratory in 1920. Fabric and cord tires of several different makes were compared and showed a satisfactory agreement in rolling friction. Tests of cord tires of this size have shown a constant decrease of friction, amounting to about 10 per cent.

The 33 x 4½-in. tires of 1919 cited in Table 14 were characterized by a low rolling-friction as shown by Fig. 12. This feature seems to have been lost since that time, and recent tests show that a tire of this size now has more rolling-friction than a 32 x 4-in. size.

Comparisons of tire tests made several years apart require proof that the differences really exist in the tires and not in the apparatus itself. It can be said that at least during the last 4 years, while the comparisons have been in progress, cord-tire friction appears to have decreased noticeably. It is noteworthy that tires made by several of the leading manufacturers have shown marked uniform rolling resistances. This indicates, possibly, that the processes of tire manufacture have become standardized.

Deviation from this uniformity has been observed in a few instances. In one case, a pair of tires marked "Cord" was found to have the resistance usually found in fabric tires, that is, 50 per cent in excess of cord tires. In two other cases, cord tires showed 27 and 30-per cent excess over the standard cord-tire figures. All the examples that deviated from the standard values were found in the products of the smaller and less well-known manufacturers.

Temperature and Heat

That rubber tires get warm while running is well known. The true source of the heat is now known to be within the tire structure. It may be due in part to compression of the rubber or in part to the flexure of the carcass or both. Heat generation is not confined to pneumatic tires but is found in the same degree in those of solid-rubber. This proves that compression of the rubber unaided by flexure of the tire walls will result in heat.

TABLE 14—ROLLING-RESISTANCE OF 33 X 4½-IN. PNEUMATIC TIRES

Name	Number	Description	Year	Inflation-Pressure, Lb. per Sq. In	LOAD, LB. 700	1,000	1,300
Fisk	59	Fabric, plain Weight, lb. 20.0 Diameter, in. 33.2	1919	30 45 65 90	13.0 10.5 9.5 7.7	21.0 16.5 15.0 12.5	28.7 22.7 20.5 17.0
Fisk	60	Fabric, non-skid Weight, lb. 23.5 Diameter, in. 33.7	1919	30 45 65 90	12.0 10.5 9.5 8.5	20.0 17.0 15.5 13.8	28.0 23.5 21.5 19.0
Firestone	55	Cord, ribbed Weight, lb. 30½ Diameter, in. 34.8	1919	30 45 65 90	7.5 6.0 5.3 5.0	12.3 9.6 8.5 8.0	17.0 13.3 11.7 11.0
Firestone	54	Cord, non-skid Weight, lb. 30.5 Diameter, in. 34.9	1919	30 45 65 90	7.5 7.0 6.1 5.5	12.0 11.0 9.8 8.7	16.5 15.0 13.3 11.9
Goodyear	68	Cord, ribbed Weight, lb. 26.5 Diameter, in. 34.1	1919	30 45 65 90	7.6 6.0 4.8 4.3	12.8 10.5 8.0 7.3	18.0 15.0 11.5 10.5
Goodyear	56	Cord, non-skid	1919	30 45 65 90	9.2 7.3 6.5 5.7	13.7 11.0 10.0 9.2	18.3 14.7 13.4 12.5
Fisk	61	Cord, ribbed Weight, lb. 24.0 Diameter, in. 33.6	1919	30 45 65 90	6.7 6.2 5.6 5.3	11.3 9.5 8.5 8.0	17.5 14.0 11.9 10.9
Fisk	65	Cord, non-skid	1919	30 45 65 90	7.5 6.3 5.5 5.3	13.0 10.3 8.7 8.5	18.7 14.0 11.7 11.4
Fisk	72	Cord, non-skid Weight, lb. 27.0 Diameter, in. 34.4	1922	30 45 65 90	10.5 8.8 7.7 7.3	16.0 13.0 11.5 10.7	25.5 19.2 17.3 16.0
Fisk	73	Cord, non-skid Weight, lb. 29.5 Diameter, in. 34.4	1922	30 45 65 90	11.5 9.0 7.7 6.5	16.5 13.4 10.8 9.5	22.5 18.0 16.6 14.8

The flexure of a tire can be likened to the bending and unbending of annealed wire, where motion in any direction absorbs work and generates heat within the material. An ideal tire structure would be one of perfect elasticity, where the work of flexure or compression would be completely returned after the original positions were resumed. Present tire structures are at least partly elastic as indicated by the fact that cord tires have only two-thirds the rolling resistance of fabric tires; hence they have only two-thirds the heat-generating capacity.

It can be assumed fairly that the generation of heat and the resulting rise of temperature are directly associated with the rolling resistance of the tire. It can be assumed also that the amount of heat generated is exactly equivalent to the work of the rolling resistance and can be computed from this work, as in the case of the friction brake. This suggests a simple but practical way of testing the rolling resistance of two tires, which is to place them on the opposite wheels of a car and observe the surface temperatures after a long run. Measurements of the surface temperature are unsatisfactory as a final measure of tire resistance, although they may have some value as a check.

The surface temperature can be computed, approximately, from the rolling resistance. Useful conclusions can be drawn from such computations. It can be shown, for instance, that a large-size pneumatic-tire will heat more than a small one. The heat equivalent of the rolling resistance in British thermal units per hour can be found by obtaining the product of the resistance times the speed times 2546/375. The heat dissipated can be expressed as the product of the cooling surface or area times the difference of temperature between this surface and the surrounding air times a coefficient. The coefficient of heat transfer is fairly well known from experiments on moving air over surfaces, such as hot-blast heaters, automobile radiators and the like.

Applying this method to a 32 x 4-in. cord tire at a speed of 30 m.p.h., a computed rise of temperature of 29 deg. fahr. was obtained, using a normal full load of 1000 lb. Applying the method to a 44 x 10-in. cord tire at a speed of 30 m.p.h., with a load of 7000 lb., a computed rise of temperature of 90 deg. fahr. was obtained. This increase of temperature is due to the fact that the load capacity and the resultant heat of tire-friction increased faster than the surface for heat dissipation. The computed temperature-rise of small-size tires has been satisfactorily checked on the Mason Laboratory drums, but thus far no observations have been made on the large sizes.

Gearbox Lubricant

The transmission friction may be materially affected by the temperature of the lubricant in the gearbox. This is true especially of trucks, where the transmission friction normally is large. Tests were made by leaving the truck exposed all night to November weather, then running the rear end on the drums and making friction measurements at intervals as the lubricant warmed-up. Runs were continued until the lubricant temperature had risen from 35 deg. fahr. to about 135 deg. fahr.

Fig. 15 shows the observed rise of temperature and the drop in the rear-end resistance of three trucks. In two cases the rear-end friction was increased 50 per cent by the coldest lubricant. The effect of friction is to produce heat that warms the gearbox and its contents, thus automatically reducing the friction. A temperature of 140 deg. fahr. in the gearbox was observed that was produced solely by internal friction. No attention was paid to the gearbox temperatures of passenger cars. All tests were made at the ordinary room-temperature with the engine

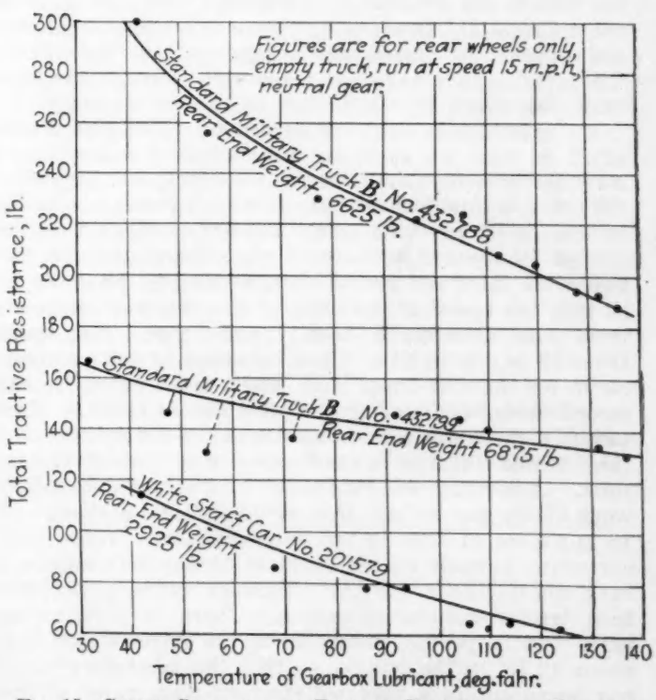

FIG. 15—CURVES SHOWING THE TRACTIVE RESISTANCE OF TRUCKS AT DIFFERENT TEMPERATURES OF THE GEARBOX LUBRICANT

and the transmission well warmed by preliminary running.

THE DISCUSSION

MAJOR M. L. IRELAND:—I would like to express the appreciation of the engineering division of the National Research Council for the very cordial cooperation that we received from Yale University in this work. We doubled the greatest pulling load that they had ever had on this apparatus. The maximum force at the tire surface illustrated by Professor Lockwood was about 600 lb. When we put the 7½-ton Mack truck on it, the pull ran to 1100 lb. The Quartermaster Corps' standardized Class B truck made it 1000 lb. The work done by Professor Lockwood was an essential part of our research because we were endeavoring to tear apart the group of forces that enter into what is known as tractive resistance, and it appears from a very careful survey that we have made of the literature of Western Europe and America on the subject of tractive resistance that there has been no really successful effort to separate that group of forces into its component parts; although Riedler has separated laboratory results, as distinguished from road results, into component parts. It is only by some such method as Professor Lockwood's, particularly when we are dealing with the gasoline-driven vehicle, that we can get at these results. When these results are published, the first thing that engineers will say is, "Why did you not run the speed test out to a greater range?" The answer will be that we went to the limit and beyond the capacity of the equipment. A survey of the equipment installed elsewhere indicates that no one has equipment that will withstand these big heavy trucks under full load and at high speed. One of the developments that I think the next few years will bring is the redesigning and rebuilding of apparatus similar to Professor Lockwood's to handle these heavy loads. Professor Lockwood is leading the way in a creditable manner.

Many of the best researches conducted in the past have wholly overlooked the matter of the degree to which the results are affected by the change of temperature in the differential. In scanning two or three researches that are regarded as practically the last word, we found that the investigators have not taken this into account, and that therefore their results may be affected seriously.

An interesting result in regard to this temperature-effect is this: we made six runs about 4 miles out of New Haven with the Mack 7½-ton truck, one run after the other, a total elapsed time of 82 min., with absolutely no change of any kind in the conditions other than the change in the temperature of the differential and, between the third and fourth runs, we opened the governor so that the speed at the crest of the hill was increased from about 11 m.p.h. to about 17, and the maximum speed from 18 m.p.h. to 21. When we came to take a mean curve for the first three runs, and a mean curve for the second three, we found altogether different results. Fortunately, there was a thermometer in the differential case that showed a change in the temperature of about 30 deg. fahr. According to Professor Lockwood's laboratory work of the day before, this would indicate a change of .15 m.p.h. or 11 tons in tractive resistance. When that correction is made on the chart at 15 m.p.h. it accounts very satisfactorily for the difference between the two final tractive-resistance curves. There is perhaps as much as 9 lb. per ton difference at the extreme ends and about 12 lb. in the middle, so that the laboratory work has borne out in a very satisfactory way many of the results that we are obtaining by actual measurement on the road.

The work of measuring tractive resistance covers a wide range of tires running over a wide range of loads. We have made tests on five different kinds of road thus far.

HERBERT CHASE: — Professor Lockwood's excellent paper represents an enormous amount of work. I found much useful information in the data that he has accumulated and prepared an article entitled Rear Wheel Dynamometer Tests and Their Significance to the Engineer.[3] I had an opportunity to pick out from some 200 tests that Professor Lockwood made, about 20 tests which seem to be fairly representative of the passenger-car field in particular and include also three or four trucks. The data that I secured are tabulated in the article referred to and the curves drawn for comparative purposes are shown therein.

The figures on wind-resistance in the table are computed, as already explained, by determining the projected frontal area of the car and multiplying the number of square feet by 0.003 and the square of the speed in miles per hour. This, of course, is only an approximation since the wind-resistance is affected materially by the streamlines of the body, but it serves well enough for comparative purposes. According to the figures given regarding these 15 to 20 passenger cars, the frontal area varies from 16 to 32 sq. ft. in the passenger cars listed. These areas no doubt could be decreased in some cases by improved body-design. Some means of actually measuring wind-resistance is needed before the actual resistance of certain other types and sizes can be predicted with certainty. Wind-resistance becomes an important factor at speeds as low as 20 m.p.h.

The method of obtaining figures for tractive effort on level roads used by Professor Lockwood is to add to the wind-resistance the resistance of the front wheels of the car. This gives the pull that must be exerted at the periphery of the driving-wheels to propel the vehicle on a hard, level road at the various speeds indicated.

In testing cars on the dynamometer, the brake is set to give loads equivalent to the tractive resistance on the level road at the respective speeds, and the fuel-consumption is measured. The two factors that oppose the forward motion of a car on a level road are the wind-resistance and the rolling-resistance of the front wheels, the rear wheels and the elements connected to them.

The power delivered by the rear tires is the power that remains after overcoming the losses from the engine back to the rear wheels including the power lost in the rear tires, and that is precisely equal to W, the wind-resistance, plus R, the front-wheel resistance; R can be measured accurately by the method Professor Lockwood has described. The value of W is computed from the formula mentioned and is only an approximation. I suggest a more accurate method of obtaining the factor W. It is a method which will take into account the streamline form of the body; in short, measure its actual resistance. Quoting from my article

> The brake horsepower that the engine is required to develop for level-road operation at average driving speeds is small compared to its maximum power at these speeds. The average at 20 m.p.h. for 17 passenger cars is about 5.4 hp. as against a maximum average at the same speed of 23.8 hp. In other words, under what can be considered a normal operating condition, the average engine develops from one-fifth to one-quarter of the power it is capable of developing at that speed. Even the heaviest of the passenger cars tested requires only 7.55 b. hp. to propel it at 20 m.p.h. on a level road,

[3] See *Automotive Industries*, April 20, 1922, p. 859.

while some of the lighter cars require only a little over 4 b. hp. It seems almost incredible that if facts such as these had been more generally appreciated, so little attention would have been given to making engines more efficient at the light loads obtained in normal use.

It is instructive to note from the tabular data the relative importance of the three losses, which together exactly balance the power developed at the flywheel of the engine. These are (*a*) the rolling-resistance of the front wheel, (*b*) the power losses between the engine and the driving surface of the rear tires, including the losses in these tires, and (*c*) the wind-resistance.

The method of measuring the brake horsepower, in the test that Professor Lockwood makes, is to drive the rear wheels from the drums to which power is applied by the electric motor below the floor. The power is transmitted through the tires and the bevel gears to the upper shaft of the transmission. The gears are placed in neutral. There is some loss due to the churning of the lubricant in the gearcase. There is also a loss in the gearing, which may or may not be the same as that which occurs when the engine is driving the car under load. In other words, the gears are running light as compared to running under load when driving the car. Furthermore, there probably is a certain difference between the power absorbed by tires that are merely rolling and tires that are driving. For these reasons the power losses thus secured, when added to the rear-wheel horsepower, probably do not give exactly the brake horsepower of the engine, but for comparative purposes they are doubtless good enough. If Professor Lockwood had the equipment, it would be interesting to know just how much greater this loss would become if the gears and parts that transmit the power were actually loaded.

In regard to wind-resistance, a suggestion is made in reference to a method of measurement, more accurate than using the computation based on the frontal area. To do this the car should be equipped with a vacuum gage, connected to the inlet-manifold, and an air-speed meter. It would then be taken to a stretch of level road and operated at the desired speeds, checked by speedometer and stopwatch. In each case the reading of the air-speed meter and the vacuum gage would be noted, the former indicating any head or tail wind that might affect the results. The car would then be brought back to the laboratory and tested at the same speeds and throttle positions, as indicated by manifold depression, and the rear-wheel power measured. The difference between this rear-wheel power and the power absorbed by the front wheels would be the actual power used in overcoming wind-resistance at the respective speeds.

To secure a more certain determination of the brake horsepower, and consequently a more accurate method of measuring the losses between the engine and the rear wheels, requires equipment that may not now be available in the Mason Laboratory and that may not be as easily connected up and used as the present equipment. I am under the impression that it would be possible in some cases to make a connection to the front end of the engine in the place of the starting-crank, and by a light prony brake, or some other form of brake, measure the brake horsepower of the engine directly. Measuring the difference between the brake horsepower and the rear-wheel horsepower, as it is now measured, would afford a means for precise determination of the losses between the engine and the periphery of the rear tires.

The losses in the rear tires can probably be assumed to be approximately equal to the losses in the front tires, if the tires themselves, the inflation-pressures and the weights on the tires are the same, as they would be for purposes of calibration.

E. FAVARY:—Will Professor Lockwood give particulars regarding the difficulties he experienced with the drawbar-pull tests of his earlier dynamometer?

As to the losses in tires, some of the results I found a few years ago probably hold good at present. Professor Lockwood showed that a partly worn solid tire had less rolling-resistance than a new solid-rubber tire. This is easy of explanation when we consider that the tests are made on a smooth drum. If the road surface were perfectly smooth, a solid-steel tire would, I think, show the least resistance. However, the rougher the road is, the softer must be the pneumatic tire to show the least rolling-resistance, because the most perfect tire is that which will absorb the average height of obstructions on the road without raising the axles. The solid tire has an increased loss on account of the low elastic efficiency of rubber.

PROF. E. H. LOCKWOOD:—Mr. Favary has raised an interesting question on the relative resistance of solid-rubber and pneumatic tires on smooth and on rough surfaces. His doubts that solid-rubber tires on rough roads would roll as easily as fabric pneumatic tires seem well justified. Experiments are now in progress in three different States for road resistance of loaded trucks on smooth portland-cement roads. Preliminary figures from all of the tests appear to indicate that solid-rubber tires have actually lower rolling-resistance than fabric tires on passenger cars. Data are not at hand for badly worn roads, where doubtless the solid-rubber tires would offer relatively more resistance.

THOMAS S. KEMBLE:—Has Professor Lockwood been able to determine the average overall efficiency of passenger-cars and trucks?

PROFESSOR LOCKWOOD:—Questions about efficiency are difficult to answer for the simple reason that the efficiency of a vehicle has not been defined. I should therefore begin by defining the term as the ratio of the power delivered at the periphery of the rear tires to the power developed at the clutch. According to this definition, if the friction losses between the clutch and the rear tires could be reduced to zero, the efficiency would be 100 per cent. On the other hand, if the engine is idling with gears in neutral, the efficiency will be zero, because in that case no power is delivered to the rear tires. Average efficiencies under normal operating conditions can be computed for typical vehicles, as shown in the several

TABLE 15—AVERAGE MOTOR-VEHICLE EFFICIENCIES UNDER NORMAL OPERATING CONDITIONS

Vehicle	Kind of Tire	Tire Size, In.	Weight, Empty, Lb.	Speed, M.P.H.	Efficiency, Per Cent	
					Level Road	Steepest Grade
Light Touring Car—Dodge.....	Cord	32x4	2,655	20	41÷ (41+ 25) =62	295÷(295+ 30) =91
Heavy Touring Car—Cadillac.....	Cord	35x5	4,395	20	61÷ (61+ 56) =52	577÷(577+ 53) =90
Heavy Truck—Standard, Class B..	Solid Rubber	....	11,540	15	111÷(111+127) =47	588÷(588+135) =81

examples chosen from tests at the Mason Laboratory that are given in Table 15.

It appears that the efficiency as defined may extend from zero to a maximum of 90 per cent. It is gratifying to see that an efficiency of 90 per cent can be realized, but it must be remembered that under ordinary running conditions the figure is much lower, say 50 to 60 per cent.

MR. KEMBLE:—Did you give the tractive resistance of the front wheels separately from the rear wheels? Is your comparison of the cord, fabric and solid tires for the front or for the rear wheels? It seemed to me that it was for the rear wheels and, unless there is some further explanation, we might be led astray. There are transmission losses which, if included, affect the percentages; whereas, if the comparison were made on the front wheels, the loss in the bearings being comparatively small, the percentages would be more accurate.

PROFESSOR LOCKWOOD:—That is an important point. When we measured the frictional resistance of pneumatic tires, whether on the front or rear wheels, we invariably got the same results. Ordinarily, the front-wheel bearing-resistance is from 4 to 6 lb.; the rear, including transmissions, from 15 to 20 lb. This resistance, subtracted from the total, leaves the difference we call tire resistance. We believe that the tire resistance is the same whether measured on the front or the rear wheels. Our tests have been made on the rear wheels, as a matter of convenience in changing the loads.

MR. KEMBLE:—You take as the tire resistance of the rear end the difference between the resistance with the load on and the resistance when the wheels have been raised so that they just touch the drum; has that been checked to a certain extent with the resistance that you get on the front wheels?

PROFESSOR LOCKWOOD:—Yes.

C. F. SCOTT:—Is the large-diameter drum essential to successful results? Could satisfactory results be expected with, say, 3½-ft. drums instead of 5-ft.; and with steel drums?

PROFESSOR LOCKWOOD:—I do not feel very positive on this point. Whether it could be predicted that the small-size drum would give different results, I am not prepared to say. It would be an interesting experiment to have the same vehicle tested on the steel drums at Worcester, and then on the Mason Laboratory drums, to see whether the results in one place would be the same as those in the other. When Mr. Favary's apparatus at Cooper Union, New York City, is completed, it would be instructive to have it compared with the others.

MR. SCOTT:—To repeat Mr. Favary's question, why did not the drawbar measurements work out?

PROFESSOR LOCKWOOD:—Like most experimenters, we tried to measure this resistance directly. The first thing we discovered was that the front wheels, resting on the floor, varied the drawbar pull by from 15 to 20 lb. To avoid this error it would be necessary to raise the front tires, allowing the front end to roll on anti-friction rollers, as was done by Riedler in Germany.

We tried to suspend the front end with our overhead crane but found difficulty with side-sway and abandoned the trial. Another thing is the difficulty of keeping the contact of the rear wheels directly over the center of the drums. If it gets slightly off-center, there is a gravity component which has some effect. Due to such troubles, we discarded the attempt to measure the drawbar pull directly and used the friction-brake pull instead.

R. E. PLIMPTON:—How much would the frictional resistance increase with the diameter of the rear wheel, if the load were constant; say with tires between 34 and 40 in.?

PROFESSOR LOCKWOOD:—It is my offhand impression that it would not make much difference. I do not know of any experiments to determine the effect of tire diameter on the rolling-resistance on smooth roads. I suspect that the effect produced by any change of the diameter is very slight, perhaps too small to measure.

CORNELIUS T. MYERS:—The old subject of the gasoline engine working at low efficiency in passenger cars and trucks is still with us and probably will be for some time. In 1910 and 1911, in the design of a new line of trucks, it was very thoroughly threshed out in one instance, and an effort was made to impose on the sales department a line of trucks with much smaller engines than had been considered good practice up to that time. We encountered considerable "rolling-resistance" in the sales department, but a tractive-factor formula was developed[4] which took into consideration the various factors that Professor Lockwood has set forth.

In this tractive-factor formula the resistance was measured in fractions of a pound per pound of weight. Professor Lockwood has suggested that instead of using pounds per ton, we use pounds per thousand pounds. I should like to go a little farther and use a plain pound coefficient. It will be a decimal of a pound, of course, per pound of weight. That is directly comparable with the grade coefficient as say 2 per cent, or 0.02 lb. per lb.

PROFESSOR LOCKWOOD:—It seems to me 100 would be more logical. Then 10 lb. per M would be 1 lb. per 100 lb. Why not take 100?

MR. MYERS:—For the reason just stated. In going into the subject I found nothing whatever in this Country, but in France and Italy the road coefficient used is kilograms of resistance per kilogram of weight. It is understandable and is the ordinary method of expressing a coefficient.

The wind-resistance for any car or truck can be determined indirectly. With how great accuracy I cannot say because I have not made enough experiments. In looking at Professor Lockwood's illustration, we see that if the car were allowed to coast, after it reached a certain speed there would be the resistance of the front and rear wheels and of the axle and transmission, if the gear is in neutral and the engine shut-off.

Professor Lockwood has obtained what we have had heretofore only approximately from Riedler's experiments. He has given us the resistance of the tires. One thing is left and that is the wind-resistance. You can get that from the accelerometer readings, from which the above resistances must be deducted. When the car is run in opposite directions, with and against the wind, if the gear is placed in neutral and the other four resistances are deducted, the wind-resistance can be determined. It is a definite and understandable figure. In 1912 and 1913 we checked the rolling-resistance of cars with the accelerometer and found, as Professor Lockwood did, that if the truck is in good condition, the rolling-resistance is about 40 lb. per ton and that of the passenger car is about 30 lb. per ton at 15 m.p.h.

M. C. HORINE:—We have talked about the mechanical efficiency of the vehicle, the wind-resistance and the tire-resistance. There is another factor, the road-resistance. Professor Lockwood's tests were performed under ideal conditions so far as the road is concerned. Still another is the friction generated by the spring-suspension of the vehicle. It takes power to lift a vehicle bodily into the air and let it down again. That is the acceleration and

[4] See S. A. E. HANDBOOK, vol. 2, p. 67.

deceleration of the mass. There is friction in the springs and the spring-shackles. It takes power to move those parts, although the power is not applied directly. The motion of the parts and the wear that results cause the consumption of driving power. Consequently, more than a laboratory test is needed to determine the exact amount of power required to move a car over a given road. Has Professor Lockwood considered the matter of springs in the power losses in the suspension of the vehicle?

PROFESSOR LOCKWOOD:—That is a topic which is rather new to me. I am not sure that it would be deserving of attention. The measurement of it would be a nice problem.

MAJOR IRELAND:—We are getting some interesting results from road tests, but it is early yet to give definite predictions, much less laws and formulas. One interesting test that we happened to make was on some granite blocks between street-car tracks down a certain hill. We did not know at the time that we had crossed the line between the cities of Everett and Malden, Mass. In Everett the blocks have cement grouting between the joints and the road surface is smooth; on the other side of the city line in Malden, the joints are sand-filled. There is a marked difference in the way some of these granite-block sets stand above one another in the two cities, and there is a marked difference in the resistances at the same velocity in two parts of the run. We have photographed several sections of the road and measured the degree of roughness with a 10-ft. straight edge.

If the amount of energy absorbed in the spring-suspension, frame, tires and other vehicle parts is subtracted, there would be included in the remainder that part of the road-test results which is due to the road itself, and this would show in a much clearer way the comparative effects of the sand-filled and grout-filled joints. The separation of the tire-resistance and the resistance of the transmission system is not difficult. How to separate the effect of the increased impact absorbed by the vehicle parts from that which is absorbed in the road material and therefore belongs in what is called road resistance is a problem. About the only means of doing this simply seems to be to deduct very carefully all tire and transmission-system resistance that is due to other causes, and then, from the principle that action and reaction are equal but opposite in direction, divide the remainder, which should be impact effect, into one-half absorbed by the vehicle parts and one-half absorbed by the road material. It will be seen that, in practice, some serious difficulties will be encountered in doing this.

AUSTIN M. WOLF:—The results obtained in these tests would be comparable to an ideal roadway. Besides the energy expended in the inter-leaf friction and the friction in the spring-shackles, much energy is absorbed in the moving of the great mass represented by the weight on the springs. The universal-joints are working at greater and irregular angularities; the tires are flexed to a greater extent than on the dynamometer drum; the air within the tires is compressed whenever a bump is met; and, in addition, there is the side-slipping and the waste of energy caused by it. After striking an obstacle, one wheel that is in the air is speeded up through the differential and when the tire returns to the ground the tread is chafed. All these factors may be small severally, but collectively they represent a considerable amount of energy.

Taking these things into consideration, Mr. Chase's method of measuring air-resistance would give a result too high, as the losses indicated on the dynamometer would be less than those actually encountered on the road.

I suggest that in future tests an attempt be made to use a dynamometer drum having a detachable tread so that various irregularities in the road could be represented. This would be somewhat similar to the method used some time ago in attempting to run a destruction test on aluminum wheels. It would be interesting to have each rear wheel rest on an independent drum and the speeds varied to reproduce the action of a car in turning a curve. The losses due to the differential might thus be determined.

It probably would be difficult to make accurate measurements on a dynamometer with a detachable track on the drums, but I believe that a careful analysis of the problem would result in adequate equipment. To get more accurate data in road tests, it occurred to me long ago that we are not measuring the development and expenditure of energy by a simple and direct means. I suggest that the engine be cradled in the chassis and the torque reaction be measured directly in this way; similarly, with a separately mounted transmission it also can be cradled and the torque reactions noted, when in other than high gear. With a torque-arm attached to the rear axle, the reaction at its point of anchorage on the chassis can be measured. All these parts should work on a recording type of instrument, as it would not be practicable to take visual reading at all these points during a test.

PROFESSOR LOCKWOOD:—The point raised by Mr. Wolf that the losses indicated on the dynamometer would be less than those actually encountered on the road, seems reasonable. It is surprising, however, that for the few vehicles thus far tested on the Mason Laboratory dynamometer in comparison with new cement roads, the road-resistance appears to be somewhat lower. It must be remembered that this comparison involves the deduction of the wind-resistance from the road results, and thus far wind-resistance has been computed by a formula and is rather approximate.

The dynamometer results are less ideal than might be expected. The Mason Laboratory drums are not perfectly round. The variation of the radius at present is about 1/16 in. Due to this and variations of the tires themselves, most cars run on the drums with a slight up-and-down motion, which undoubtedly absorbs power just as on the road.

No mention has been made of the continual rolling of the tire on the curved drum surface. This is a different condition from rolling on a flat surface, and probably the curved surface will offer more resistance.

A. E. F. ARTILLERY GUNFIRE

ABOUT 2500 guns of all calibers, nearly 700 of which were heavy cannon, were with the First Army American Expeditionary Force, during the Meuse-Argonne offensive. During the battle these guns fired about 4,215,000 rounds, or a greater number of rounds than was fired by the Union forces during the entire Civil War.—*Journal of the United States Artillery.*

Electric Wiring for Automobiles

By William S. Haggott[1] Buffalo Section Paper

THE importance of the electrical wiring on automotive apparatus is indicated by the fact that, in effect, a central electric-power station, as well as its distributing lines and customers, is consolidated upon a high-speed vehicle and the system, inclusive of high-tension and low-tension circuits, direct and alternating currents, generators, transformers, motors, storage-batteries, relays, circuit-breakers and fuses, is expected to function properly every day under all conditions.

The three main divisions of electric wiring for automobiles are high-tension ignition, lighting and starting circuits. The present practice with regard to each division is discussed and suggestive comment is made with a view toward improvement, one feature worthy of serious consideration being that of the adoption of a standard color-scheme whereby the character and service of each separate circuit on any car would be indicated by the color of the wires used.

THE Standards Committee of the Society has given some consideration to the possibility of standardizing methods of installing wiring on automobiles, with the purpose of adding to the usefulness of the existing S.A.E. Standard for Insulated Cable,[2] covering the construction of individual cable. Within certain limits it seems feasible to do something along this line, and it is hoped that this paper and its discussion will serve as a basis for further efforts. I am indebted to various members of the Electrical Equipment Division for much of the information I have used in preparing the paper. Their cooperation has been of great assistance.

We are all more or less familiar with industrial electric powerplant and transmission-line design and construction. We also know that at the receiving end of the transmission line we have energy which is available for light, heat and power. However, the average person seldom stops to think that on present-day automobiles we have practically all of the electrical features that are found in every zone served by a central station. We have high-tension and low-tension circuits; direct and alternating currents; generators, transformers and motors; storage-batteries, relays, circuit-breakers and fuses. In fact, we have the central station as well as its distributing lines and customers all consolidated upon a high-speed vehicle, and the system is expected to function properly every day, in all sorts of weather and under all conditions.

With the foregoing thought in mind, even the layman must readily see the importance of the electrical wiring on automotive apparatus. A broken wire, a loose terminal, a short-circuit or any one of a number of things, can cause delay, inconvenience and often danger. However, automobile cable construction and its installation cannot be patterned after regular light-and-power cable construction and installation. The conditions under which automotive cable does its work are entirely different from those imposed upon light and power cables. Automobile wiring falls naturally into the three main divisions of high-tension ignition; lighting, including low-tension ignition; and starting circuits.

High-Tension Ignition

Present practice and opinion are about evenly divided as between plain rubber-covered and braided rubber-covered high-tension cable. However, I believe the tendency is strongly toward the braided construction, provided the braid is thoroughly filled with a durable flexible insulating varnish. The principal objection to braided cable is that the braid retains moisture, thereby creating a possibility of erratic firing, but this objection can be overcome if the braid is properly treated, and considerable progress is being made in this direction. Braided cable, properly constructed, affords protection from oil, heat and ozone, all of which seriously damage a plain rubber-covered cable. Ozone is the worst of the three, as it causes the rubber to crack and results in misfiring or irregular firing.

If the high-tension wires are short and can be kept free from injury, it is preferable to run them open and not close together. This method gives positive protection against inductive disturbances and leakage between the wires, and will prevent "cross-firing." I believe that in many cases the spark-plug wires can be run open to good advantage, but there are instances, especially in connection with 6, 8, and 12-cylinder engines, some of which have two spark-plugs per cylinder, where it becomes almost and perhaps actually imperative to run the wires in tubes.

Metal tubes are usually employed for this purpose on account of cost. A fiber tube is better, as it will prevent leakage to the ground in case the cable insulation fails. To nullify the condenser action when a fiber tube is used, it is only necessary to run a bare wire through the tube and ground this wire to the engine. Where a metal tube is used, it must be well grounded to the engine so as to carry any static or condenser effect to the ground. The openings in a metal tube must be properly rounded or bushed to prevent cutting the cable insulation. If possible, the coil should be located close to the distributor in battery-ignition systems, thereby keeping the wire from the coil to the distributor short. Where a tube is used for the spark-plug wires, it is best to keep the wire from the coil to the distributor out of the tube.

Lighting and Low-Tension Ignition

There is a strong tendency toward the use of armored cable on lighting circuits. This is particularly true where wires are exposed and subject to chafing. Armored cable, if properly constructed and installed, will give no trouble. The most important factor in the use of armored cable is to be sure that the armor is stripped back sufficiently from the terminals and then soldered down so that a ground cannot occur at these points.

Rubber-covered-and-braided or varnished-cambric-and-braided cable protected by flexible metallic conduit is sometimes preferred. There is also some use of non-metallic conduit to carry these types of wire. The use of metallic conduit perhaps gives a more substantial appearance on head and tail-lamp wires, but is more expensive

[1] M.S.A.E.—Cable sales manager, Packard Electric Co., Warren Ohio.

[2] See S.A.E. Handbook, vol. 1, p. B 33.

than armored cable in both first cost and installation.

It is unnecessary, of course, to have metallic protection for wiring around the instrument-board. I have in mind several installations in which rubber-covered-and-braided cables have been used for years on all circuits, including those for head and tail-lamps, without the use of additional protection, but I believe the use of armored cable is good insurance against lighting-circuit failures.

The question came up recently as to the advisability of running separate head-lamp feed-wires from each head-lamp clear back to the lighting switch, instead of making a splice at a point near the front end of the car and having only one wire from the switch to the point of the splice. The latter method has practically become standard. The splice can be properly made and supported and, when so constructed, will give no trouble. It is better not to wire instrument-lamps in series with tail-lamps as is sometimes done. The only advantage in the series arrangement is to indicate when the tail-lamp is burned-out, and this fact can be checked up easily by looking back for the reflection of the tail-lamp. With tail and instrument-lamps in series, it is necessary to use 3-volt bulbs, which are rather special and sometimes hard to obtain. The series arrangement also requires continuous burning of the instrument-lamp, which is fatiguing to the driver's eyes. With separately controlled lamps, the instrument-lamp can be turned out when the car is parked, and can be substituted for the tail-lamp if it should burn out.

Where fuses are used, each circuit should be protected, even though this is slightly more expensive than to have one fuse control several circuits. With each circuit fused separately it is easier to locate trouble and, when circumstances require, it is possible to operate the car with one circuit out of commission. Considerable attention is being given to the use of a single circuit-breaker placed in the main battery-feed circuit. This method of protection has the advantage that, when the cause of trouble is removed, the circuit-breaker can be closed and the car can proceed without resorting to the installation of make-shift fuses, which is always dangerous.

It is good practice to consolidate lighting and low-tension wires wherever possible and to braid them together or run several wires through one piece of conduit. This method is economical and convenient and tends toward rugged and safe installation. If the proper wire is used and the individual wires are assembled in a workmanlike manner, installations of this sort give little or no trouble from the repair-shop standpoint. This use of assemblies, or wiring harnesses, as they are often called, is admirable from a car-assembly standpoint. The wiring harnesses are formed-up on a bench, which permits uniformity and careful workmanship and a more satisfactory wiring job on the car.

I suggest that some effort be made to adopt a standard color-scheme for various circuits on all cars. For example, every car has a wire running from the battery or starting-switch to the ammeter. Is there any reason certain definite colors cannot be assigned to this and other wires that are common to all cars, so as to have as nearly as possible a standard wiring color-scheme among all car builders? It seems that this would be of great value to the car driver and repairman and provide a uniformity that would be helpful to the industry in many ways.

STARTING

The most general practice is to use rubber-covered-and-braided cable. On 6-volt systems, No. 1 American wire gage wire is the prevailing size, although considerable No. 0, as well as some No. 2, is used. The gage size must be right in order that proper starting-torque can be delivered by the starting motor in cold weather. Starting-motor cables should in all cases be as short as possible, to keep the voltage-drop to a minimum.

If the starting-motor cables are short and well supported, they will in many cases need no additional protection such as conduit or armor. Non-metallic conduit affords excellent protection against chafing or mechanical injury. If metallic conduit is used, great care must be taken to prevent the conduit from coming into contact with the terminals. This same precaution must be observed if an armored starting-motor cable is installed.

A most important point in connection with starting-motor cables is that the battery be well grounded. I believe a good standard method would be to use a two-hole lug that can be bolted firmly to the frame.

It is impossible to exercise too much care in attaching and soldering terminals. Wherever spade-type terminals are used, they should have two sets of wings, one pair to grip the insulation and the other pair to grip the copper core. I am much in favor of soldering terminals whereever possible, thus insuring proper electrical connection. There are some types of terminal and lamp connector that cannot be soldered, but I would like to see the solderless types used only in cases where soldered terminals are not at all feasible.

GENERAL

I believe it is possible to arrange the electrical wiring and the various parts of the gasoline-supply equipment so that no gasoline can leak onto any part of the electrical equipment that is liable to emit sparks. If this is done, one of the greatest fire-risks is eliminated.

The automobile as we know it to-day is largely indebted to the electrical industry for its popularity. The electrical equipment gives little or no trouble and its continuity of service depends entirely on the wiring system. The installation of wires is being improved constantly and it is hoped that the entire subject will receive continuing careful attention, resulting in still better electrical service on automotive apparatus.

THE DISCUSSION

N. J. FARRELL:—Why is it that the negative side of the battery is grounded on the frame on the majority of cars but that, on one very high-priced car, the positive side is grounded?

W. S. HAGGOTT:—After investigating, we found that about half of the automobile builders ground the negative and the other half ground the positive side. The S.A.E. Standard[3] is that "the storage battery should be grounded on the positive side, and by only one conductor."

E. T. MATHEWSON:—Why was the single-wire system adopted in preference to the two-wire system?

MR. HAGGOTT:—It is a matter of convenience and economy. We have found that it is a very easy matter to ground one side of the parts of our equipment. Grounding one side of the motor and generator, head-lamps and the like, reduces the cost and, in some cases, it has been found that a grounded system is more efficient electrically than a two-wire system. It lessens the amount of wire on the car by about 50 per cent.

MR. MATHEWSON:—I do not question the efficiency so much as the value of insurance in the one system as preferred to the other. It strikes me that there is much more opportunity for a short-circuit in the single-wire than in the two-wire system. Further, in the places

[3] See S.A.E. HANDBOOK, vol. 1, p. B 29.

where the wires lead to the ground, a bolt, nut, screw or something of the kind is required to make that contact complete; often, these loosen and cause a spark that is likely to be produced very close to the gasoline supply-tank itself. I have wondered whether that has any effect on the rate of insurance.

MR. HAGGOTT:—Assuming a two-wire system instead of a grounded system, the liability to loose connections is just as great in that case as it is in the grounded system, and there is practically the same sparking danger. Further, the two-wire system leads into serious ignition troubles where battery ignition is used, if short-circuits appear on either side of the lighting or starting circuits.

I would like particularly to have opinions expressed in regard to the feasibility of using a standard color-scheme for the wires of the different circuits.

DR. E. K. STRACHAN:—How far will a standard color-scheme be valuable to all the different makes of car? They will all show some variation.

MR. HAGGOTT:—That is difficult to determine; but every car has head-lamps and every car gets its battery feed in a certain way. No doubt there will be some variations, but the idea in general seems feasible. Practically all cars feed the ignition and lighting circuits from the hot side of the starter switch. That particular wire would, therefore, be easily standardized as to color. We can have a standard color for all lighting circuits, another standard color for all generator circuits and another for all ignition circuits. We do not need many colors; perhaps four or five circuits are common to all cars. Some cars do not have intricate systems of wiring. Of course, there are circuits on the high-priced cars that are not used on other cars, but they are not within the scope of what we are trying to do. I am strongly in favor of a standard color-scheme. It would be a very great convenience to the wire manufacturer, and to the automobile builder. Anything we can do to further its adoption will help the garage-men, the repair-men and the car-owners.

XEN FAGAN:—I can see a number of very good reasons why the color system should be adopted and, as a manufacturer, I am entirely in favor of the adoption of standardized colors for the electric wiring so far as this is possible.

MR. MATHEWSON:—Has Mr. Haggott found any standards as to a color scheme among the car builders with whom he has come into contact?

MR. HAGGOTT:—Some of them send out a separate wiring-diagram and specify on it the colors of wire for the various circuits, but no two companies use the same color on the same circuit.

N. S. DIAMOND:—Is the action of ozone physical or chemical?

MR. HAGGOTT:—Ozone is a result of static discharge or leakage. There is always a static effect around high-tension ignition just as is the case around a high-tension transmission-line. The effect of ozone on rubber-covered cables is chemical. The chemical action of ozone, however, results in a physical manner that shows up in the form of cracking. I have done some testing along that line. The best way to indicate the action of ozone quickly is to take a piece of high-tension cable, 2 or 3 ft. long, wind it tightly around a mandrel of $\frac{7}{8}$-in. diameter and connect one end of the cable to the terminals of a magneto. The other end of the cable is connected to one side of a spark-gap. The other side of the spark-gap is grounded and the mandrel itself is grounded. When the cable fails the spark will cease to jump the gap.

Using some pretty good cable in this sort of test, the samples failed within 20 to 25 min. We found a series of fine cracks opening upon the outside of the wire. We do not find these cracks where the rubber is under compression, but on the outside where the rubber is under tension. There are several ways of protecting cable against ozone. The best way I know of at present is to cover the cable with a braid, and then saturate the braid with varnish. You might just as well leave off the braid if it is not thoroughly impregnated with varnish. There has been some development along the line of taking a plain rubber cable and putting varnish on it, but we seem not to be able to find a varnish that will not crack when the cable is bent. If we can find such a varnish, we would not need the braid.

Automotive engineers have problems that are not common to all of the other fields of engineering industry. For instance, the wire and the cable for use in automotive apparatus should not be compared with those for wiring a building, either in the construction of the wire itself or in its installation. For that reason the S.A.E. Wire and Cable Standards have been worked out along individual lines based upon automotive experience of a number of years, as well as in a general way upon what has been done in wire construction in the electrical field at large.

R. W. A. BREWER:—It has been a custom in Europe for a long time to mold the high-tension cables into a tube. It has been thought that this has caused considerable lag in the system. I would like to know whether this is true. It seems to me that the construction is very good mechanically.

MR. HAGGOTT:—I have seen one or two installations of that kind. It is feasible from the standpoint of insulation, so far as individual conductors are concerned but from the standpoint of proper firing has, I believe, caused some trouble. Imagine a six-cylinder engine with the high-tension wires all molded closely together. There is a possibility of an inductive effect being set up in this group of wires, actuating a spark in a cylinder at the wrong time. This is called cross-firing. The same thing may occur with fiber tubing if some means is not provided for carrying the inductive effect to ground.

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Processing Spline Shafts by a New Method

By JAMES A. FORD[1]

DETROIT PRODUCTION MEETING PAPER *Illustrated with* PHOTOGRAPHS

THE process devised by the author was evolved to eliminate the difficulties incident to the finishing of the spline and body portions of a spline shaft, such as is used in transmission gearing, by grinding after the shaft has been hardened, and is the result of a series of experiments.

The accuracy of the finished shaft was the primary consideration and three other groups of important considerations are stated, as well as four specific difficulties that were expected to appear upon departure from former practice.

Illustrations are presented to show the tools used, and the method of using them is commented upon step by step. The shaft can be straightened to within 0.005 in. per ft. of being out of parallel with the true axis of the shaft, after the shaft has been hardened, and it is then re-centered true with the spline portion.

THE general practice among manufacturers of transmissions requiring a spline shaft has been, for many years, to finish the spline and body portions of the shaft by grinding after hardening, but it has been found that this process necessitates extreme care in the obtaining and maintaining of the desired form.

[1] Production department, Studebaker Corporation of America, Detroit.

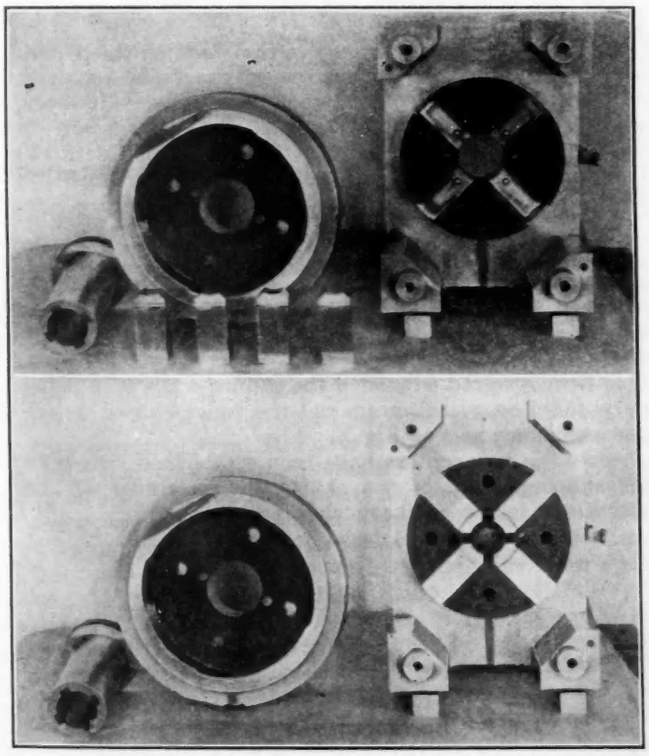


FIG. 1—TWO VIEWS OF THE DIE USED TO PRODUCE THE SPLINE SHAFTS
In the Upper View the Cutters Have Been Removed from the Die. The Cutters Are Shown in Place in the Die in the Lower View. The Cam Ring That Appears at the Left of Both Views Controls the Movement of the Cutters in Practically the Same Way That Chasers Are in a Threading Die

FIG. 2—A TYPICAL SPLINE SHAFT SHOWING HOW THE BODY OF THE SHAFT IS LARGER IN DIAMETER THAN THE PORTION BETWEEN THE SPLINES

Since it was desirable that these difficulties be eliminated, it was decided by the corporation of which I am a representative that the best method would be to omit the grinding operation and substitute a process that would not include these grinding troubles. Experiments along this line consequently were carried through by the methods and standardization department of our corporation, and the results of its investigation are presented briefly as follows.

The accuracy of the finished shaft is the primary thought that was borne in mind during the investigation, and the other important points considered were that the

(1) Splines must be straight, in line with the axis of the shaft, uniform in width, properly spaced and smooth on the wearing portion

(2) Body of the shaft between the splines must be round and parallel with the axis of the shaft, the diameters must be held within prescribed limits at any given portion and the surface must be smooth

(3) Entire shaft must be true in relation to its axis, of the proper degree of hardness and made of the

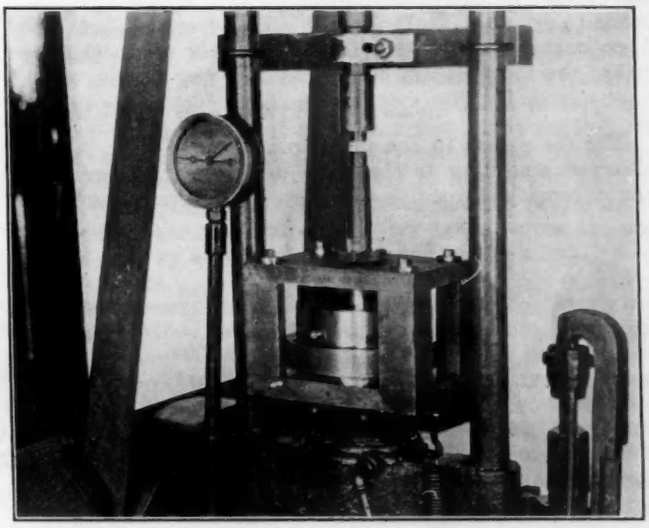


FIG. 3—A SHAFT BEING PRESSED THROUGH THE DIE TO FORM THE SPLINES

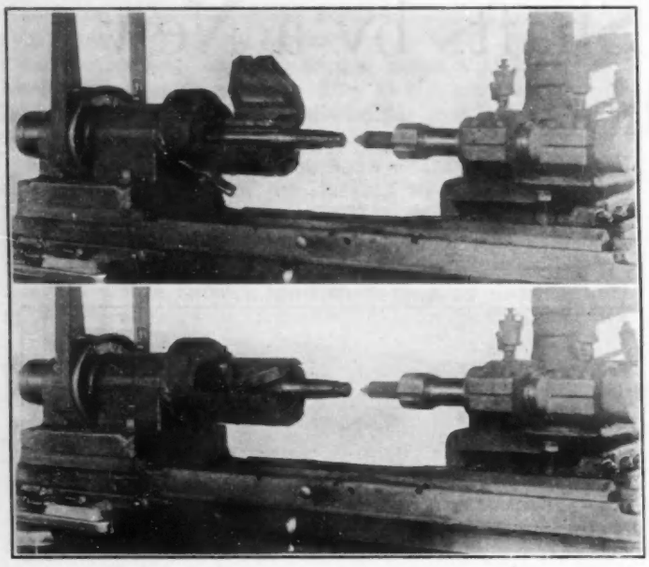


FIG. 4—GRINDING THE CENTERS IN THE SHAFT AFTER THE SPLINES HAVE BEEN FORMED

The Shaft Is Placed in the Fixture Shown in the Upper View and After the Fixture Has Been Closed and Clamped as Illustrated Underneath the Centers Are Ground with a Pencil Wheel. This Method Insures Centers That Are True with the Spline Portion of the Shaft

best obtainable material for the purpose, without regard to any difficulties that this requirement may introduce into machining operations

It was evident that other troubles would appear after departure from the grinding troubles that were experienced in the finishing of the shaft by the old process. The troubles that were anticipated were as follows:

(1) Difficulty in hobbing to exact dimensions, obtaining a smooth, even surface and maintaining a true form
(2) Liability of warpage in shafts during the process of hardening
(3) Variation in diameter due to hardening
(4) Variation in the width of the splines because of hardening

It was believed that a given allowance could be made for shrinkage during the hardening process and that a uniform shape and size could be determined in advance. Therefore, to overcome difficulty (1), the decision was made to draw the shaft through a die after the shaft had been carburized. It was found possible to do this by using the methods and tools that will now be described.

TOOLS AND METHODS

The die shown in the upper portion of Fig. 1 is constructed according to the same principles that apply to the automatic threading-dies now in general usage, the cutters being in the position that the die chasers ordinarily would occupy. In the lower portion of Fig. 1, the cutters are installed in the die. The cam ring is practically of the same construction as that used on a threading die, except that the ring is made stronger. This opening feature is necessary on account of having to pass the shaft back through the die, because the body of the shaft beyond the splines is larger than the body-portion between the splines and the shaft will not pass completely through the die. This is illustrated clearly by the view of the shaft shown in Fig. 2.

The shaft is entered into a bushing that is lined-up with the die. Then the shaft is pressed through the die to a stop that has been set at a sufficient distance to permit the shaft to pass through to the shaft-neck at the end of the splines, as illustrated by Fig. 3. The cutters in the die are then released and the shaft is removed.

During the experiments with this die, it was ascertained that a clearance angle on the cutter of 30 min. was about correct, and it was decided that one pass of the shaft through the die gave the most satisfactory result.

Up to this point we had a shaft that was carburized, and its body and splines were finished to the dimensions desired. After hardening a number of shafts, we were able to determine just what change took place in them, and an allowance was made for this in the adjustment of the die. The change was uniform to a reasonable degree in all shafts, and this enabled us to determine what the standard allowance should be.

WARPAGE

The next problem was that of overcoming the warpage in a shaft after it had been hardened. Fortunately, we found that this warpage was outside of the splined portion of the shaft.

It was very difficult to revolve the shaft on its centers and straighten it to the degree of exactness required, and the operation required too much time. However, we found it possible to straighten the shafts easily to within 0.005 in. per ft. of being out of parallel with the true axis of the shaft. Therefore, we straightened the shafts to within the 0.005-in. per ft. limit.

RE-CENTERING

After having been hardened and straightened to within the 0.005-in. per ft. limit, the shaft is gripped in a fixture mounted on the spindle of an internal-grinding machine as illustrated by the two views shown in Fig. 4. It is then revolved true with the splines while it is being re-centered by grinding against the pencil-shaped grinding-tool shown also in Fig. 4.

After the new centers have been established true with the spline portion of the shaft, the remainder of the operations follow ordinary shop practice.

The Present Status of the Air-Mail Service

By Col. E. H. Shaughnessy[1], U. S. A. Washington Section Paper

THE author outlines the history of the Air-Mail Service and states that the recent policy has been to carry out the intent of the Congress, to align the service with the desire of the administration for economy and to discontinue too rapid expansion.

After a description of the routes and divisions and a listing of the present landing-fields and radio stations, the present equipment is outlined and commented upon, tabular and statistical data being presented. The discussion covers the organization and performance of the service, the casualties, the cost of operation and the policy governing future plans.

THE Air-Mail Service operated by the Post Office Department was started in May 1918, the first step being the establishment of a mail route extending from the City of Washington to New York City via Philadelphia, no intermediate stops being made and, at first, Army planes and personnel being used. The planes were of the JN4-H type, equipped with 150-hp. Hispano-Suiza engines, and had a carrying capacity of 150 lb. of letter mail, or approximately 6000 letters. In August 1918 the Post Office Department, having completed a civilian organization, relieved the Army from further duty with the Air-Mail Service, thereupon assuming full charge of all activities and at the same time putting into use the standard E-4 type plane, equipped with the same 150-hp. Hispano-Suiza engine but having a carrying capacity of 200 lb., or approximately 8000 letters.

The Washington-New York route operated steadily and successfully. This warranted further expansion and in May 1919 the first leg of the transcontinental route was established from Cleveland to Chicago, via Bryan, Ohio, using the DH-4 type plane, equipped with the 400-hp. Liberty engine, having a carrying capacity of 350 lb., or approximately 14,000 letters. Further extension followed rapidly. Service was started in July 1919 from Cleveland to New York City via Bellefonte, Pa.; from Chicago to Omaha, via Iowa City, Iowa, in May 1920; and from Omaha to San Francisco, via North Platte, Neb., Cheyenne, Wyo., Salt Lake City, Utah, and Reno, Nev., in the following September. In addition to the transcontinental trunk-route, additional lateral routes were put into operation from St. Louis to Chicago in August 1920 and from Chicago to Minneapolis in December of the same year. When the present administration took office, there were air-mail routes in operation from New York City to San Francisco, St. Louis to Chicago and Minneapolis to Chicago, over which mail planes were flying regularly during the daylight hours.

Great things had been accomplished in the way of extending the Air-Mail Service, but the situation as we found it was not altogether satisfactory. Criticism was being directed at the service by the public and by the Congress, principally due to the fact that there had been a series of unfortunate accidents during 1920 which had resulted in a considerable loss of life. In addition to this feature, members of Congress in committee and on the floor objected to the manner in which the Air-Mail Service had been extended and financed. The whole matter was given very serious consideration. A careful check indicated that most of the operating difficulties came from too rapid expansion without providing necessary facilities with which to operate efficiently; also, there was need for a thorough understanding with the committees in the Congress.

We decided, first, to carry out the intent of the Congress, since it is not unfriendly to an Air-Mail Service but simply, and very properly, unfriendly to what it thinks is maladministration in any service. Second, we determined to align ourselves with the expressed desire of the administration for economy. Third, we planned to put into effect my own thought that we would help aviation much more effectively by stopping the too rapid expansion, which seemed to be making the Air-Mail Service an extra-hazardous undertaking through lack of sufficient facilities, and concentrated our efforts on standardizing and perfecting the operation on a more restricted scale.

For these reasons we decided to discontinue the lateral routes. That from the City of Washington to New York City was abandoned in May 1921 and those from St. Louis to Chicago and from Minneapolis to Chicago were discontinued in June 1921. This reduced our expenditures at the rate of $675,000 per annum. Today, we are operating only the New York City to San Francisco route. It is interesting to note that nevertheless our mail planes will fly at least 1,800,000 miles during the fiscal year beginning July 1, 1921, as compared with 1,760,000 miles flown during the preceding fiscal year, this showing being due to increased efficiency in flying performance.

As operated at present, the Air-Mail Service is used as an auxiliary to the fast mail-train service, because we do not attempt to fly at night. We hope that within a reasonable length of time the proposed Bureau of Aeronautics in the Department of Commerce will come into being and start the work of marking the airways for night flying. When this is done, the real value of an air-mail service will become evident at once, for with night flying mail can be put across the continent in less than 30 hr. As a matter of fact, on a test flight on Feb. 22, 1921, the Air-Mail Service put mail across from San San Francisco to New York City in 25 hr. and 21 min. of actual flying time. This was a notable performance but altogether too hazardous to try again before the way is marked by suitable aids to flying such as lighthouses and beacons.

The transcontinental route is divided into three operating divisions. The Eastern division, from New York City to Chicago, is 770 miles long, with headquarters at the former point. The Central division, from Chicago to Rock Springs, Wyo., is 1125 miles long, with headquarters at Omaha. The Western division, from Rock Springs, Wyo., to San Francisco, is 785 miles long, with headquarters at the Pacific terminal. Air-mail fields are located at Hempstead, N. Y.; Bellefonte, Pa.; Cleveland;

[1] Late second-assistant postmaster-general, Post-Office Department, City of Washington.

Bryan, Ohio; Chicago; at Maywood, Ill. (a checkerboard field); Iowa City, Iowa; Omaha; North Platte, Neb.; Cheyenne, Wyo.; Rawlins, Wyo.; Rock Springs, Wyo.; Salt Lake City; Elko, Nev.; Reno, Nev., and San Francisco. In addition there is an air-mail warehouse at Newark, N. J., and an air-mail repair-depot at Maywood, Ill.

Radio stations are located at the City of Washington headquarters and at all fields except that at Rawlins, Wyo. Navy radio stations are used jointly at Cleveland, Chicago and San Francisco. All the other radio stations are owned and operated by the Post-Office Department.

Equipment

During the period from May 15, 1918, to Oct. 31, 1921, 3 years and 5½ months, 221 planes have been used in the Air-Mail Service. Table 1 gives their history.

TABLE 1—PLANES USED FROM MAY 15, 1918, TO OCT. 31, 1921

History	Number
Crashed; Parts Salvaged	81
Crashed and Burned; No Salvage	20
Burned on Ground; No Salvage	7
Withdrawn Unsatisfactory Type; Parts Salvaged	19
Transferred to Army	1
Withdrawn on Account of Age; Parts Salvaged	8
Withdrawn and Stored; Small Type	9
Available Oct. 31, 1921, Flying Condition	50
Available Oct. 31, 1921, now Undergoing Repair	26
Total	221

Since June 1921, when the reorganization of the Air-Mail Service took place, great strides have been made in connection with standardizing and improving the flying equipment. From the first a number of different types of plane had been used. These were principally the JN4-H, the Standard E-4, the JL, the twin-engined DH-4 and the single-engined DH-4. We have standardized on the last named, eliminating all other types. This does not mean that the DH-4 is the most suitable plane available, but simply that it is a satisfactory plane for the Air-Mail Service, and that it is good business to standardize on it because the Army has a large surplus of planes of this type, which are transferred to our service, as needed, without expense to us.

In connection with the useful life of the DH-4 planes, based on our experience, Table 2 gives the record of four planes, showing what can be done in this direction. The planes are still in service and in good condition. At present we have 143 planes in the Air-Mail Service; of these 50 are in serviceable condition, 38 are being overhauled and 55 are in storage awaiting overhauling.

TABLE 2—RECORD OF FOUR AIR-MAIL SERVICE PLANES

Plane No.	Date of First Trip	Number of Miles Flown	Number of Hours Flown
71	March 3, 1919	55,416	625
99	Nov. 15, 1919	64,590	699
104	Dec. 18, 1919	58,219	634
175	Oct. 5, 1920	54,079	647

Another interesting item is that during the fiscal year ended June 30, 1921, the Post-Office Department paid $276,109 to companies for rebuilding and converting planes, and purchased eight planes for $200,000. But no planes have been purchased since July 1921 and all conversion work has been done at our own repair-shop at an average cost of $1,200. At the same time our situation has improved materially. We have 14 more planes in serviceable condition today than we had in July 1921.

Organization

The Air-Mail Service is directed by a general superintendent located in the City of Washington; he reports to the Second-Assistant Postmaster-General. The field service is divided into three operating divisions under charge of division superintendents located at Hempstead, N. Y., Omaha and San Francisco. There is an assistant superintendent for each division, as well as a local manager at each landing-field. The supply warehouse at Newark, N. J., and the repair depot at Maywood, Ill., are in charge of superintendents who report directly to headquarters. The total number of employes is 479, and the annual payroll is $787,620. Prior to the reorganization on July 1, 1921, there were 521 employes, salaries totaling $864,321.

Performance

The operating performance of the Air-Mail Service in recent months has been truly remarkable. During the first quarter of the present fiscal year 98 per cent of all scheduled trips were completed. The record, by divisions, is given in Table 3.

TABLE 3—OPERATING PERFORMANCE OF THE AIR-MAIL SERVICE

Division	Number of Miles Flown	Number of Letters Carried	Scheduled Trips Completed, Per Cent
New York City to Chicago	112,626	3,506,000	98.5
Chicago to Rock Springs	168,805	4,310,760	97.6
Rock Springs to San Francisco	109,587	2,197,520	98.0

In compiling the figures in Table 3, trips were not counted unless completed. In addition to the 391,018 miles flown with mail on regular schedule, our records show 49,662 miles of test flights and ferry trips, making a grand total of 440,680 miles flown in the 3 months. An exceptionally good record was made on that portion of the route between Cleveland and Chicago, an air-distance of 335 miles. No mail trips were defaulted from April 29 to Oct. 31, 1921, a period of 185 days. During that time 98,890 miles was flown and 3,271,360 letters were carried. Our records show that approximately two-thirds of all mail trips are made in clear weather, the remainder being in fog, rain or snow. During the same quarter there were 11 crashes, 8 of which occurred on air-mail fields and 3 at outside points due to forced landings. Two planes were destroyed entirely but during the quarter there were no fatalities or injuries to employes on regular mail trips. There was, however, one fatality on a ferry trip.

Casualties

During the period from May 15, 1918, to Nov. 30, 1921, 3 years and 6½ months, there were 30 fatal accidents in the Air-Mail Service. The record by fiscal years, from July 1 to June 30, is shown in Table 4.

During 1920, using round figures, there was one fatality for each 100,000 miles flown; since July 1, 1921, we have had one fatality for 800,000 miles flown. In my opinion this very much improved showing is due principally to putting into effect a sensible businesslike program, using trustworthy men and material and leaving the experimenting and stunting to those who are not entrusted with the United States mails. I emphasize the

TABLE 4—FATAL ACCIDENTS IN THE AIR-MAIL SERVICE

Year	Number of Months	Number of Pilots	Number of Mechanics	Number of Other Employes
1917	1.5	0	0	0
1918	12.0	2	1	0
1919	12.0	6	1	2
1920	12.0	13	4	0
1921	5.0	1	0	0
Total		22	6	2

fact that since July 1, 1921, during the time this remarkably fine record was made by our pilots, DH-4 planes were used exclusively. I cannot say too much in commendation of our air-mail pilots, several of whom have flown 100,000 miles while carrying mail. Since the beginning of the Service, air-mail pilots have flown 3,400,000 miles and carried more than 2,500,000 lb. of mail, or approximately 100,000,000 letters. No finer class of men or pilots can be found anywhere, individually or collectively, than those in the Air-Mail Service. It is a pleasure to recognize publicly the work they are doing. They are worthily carrying on the best traditions of the Service.

Cost of Operation

Statistics have been published regularly showing the operation and maintenance costs of the Air-Mail Service. They are useful for comparative purposes but have not been truly representative, in my opinion, because the figures have not included such items as general overhead expense, permanent improvements, surplus supplies on hand that were not charged out until used, planes purchased that were not charged to the capital account and the like. Using the statistics as they have been published, the purely operating and maintenance cost per mile during the quarter ended Sept. 30, 1920, was 87 cents; for the same quarter of 1921 it was 71⅓ cents, a reduction of 15⅔ cents per mile. We intend to revise the cost sheet so as to include every dollar properly chargeable to the Air-Mail Service. The new method applied to the periods just mentioned would result in a cost of $1.43 per mile for 1920 and $1.04 per mile for 1921, or a reduction of 39 cents per mile operated.

This material reduction in operating cost is chargeable to (*a*) reduced overhead expense, (*b*) increased efficiency of the personnel and (*c*) economy in purchasing supplies, particularly gasoline. We are operating successfully today on domestic aviation gasoline, whereas formerly the so-called fighting grade of gasoline was used. Cost per mile means nothing unless it is translated into ton-mile figures. Taking the last figures quoted, which are truly representative, we find that in 1921 the DH-4 plane, carrying 350 lb. of mail, was operated at the rate of $8 per ton-mile. This year, the same plane carrying the same load costs at the rate of $6 per ton-mile. We have, however, put into service recently an improved DH-4 plane with the load-carrying capacity increased to 800 lb. This makes possible an increase of over 100 per cent in utility with no additional operating cost; bringing the ton-mile cost down to $2.60. No doubt private enterprise can do much better than this, because, as is well known, the Liberty engine is expensive to operate. Our justification for using it is the surplus stock of the War Department, the first cost of it to us being nothing.

NOTES FROM A PRODUCTION NOTEBOOK

(*Concluded from p.* 412)

were tested in finished cars by inspectors who regularly passed on gear noise. The nature of the test, its object and the condition of the cars were unknown to them. Our belief in the clearance theory was again upheld by their reports. With one or two exceptions they classified the transmissions as unsatisfactory, fair or good, good or excellent, in the order of the three groups, a fact which shows that the small clearance found in some stock-bearings caused noise and that the noise was eliminated when a slightly larger clearance was provided and strictly maintained.

The amount of clearance to be allowed in any case cannot be determined by rule. We have found that it varies in transmissions similar in design but differing in detail dimensions. Our experience leads us to believe that each mounting must be studied individually, by varying the looseness of the bearing until the suitable amount is apparent. In the ball bearings referred to, a total radial clearance of from 0.0006 to 0.0009 in. was found the most satisfactory.

Backlash and Noise

From the two cases cited we judged that the same reasoning might apply in determining the proper allowable backlash in gear-teeth. Here, again, there is a possibility of not providing sufficient room for a film of lubricant, of squeezing all the oil out of the meshing space, and of causing increased noise. Experiments with gears identical in every respect except in the amount of backlash showed that a very evident drop in the noise of the gears was produced when the backlash was increased to a certain point. We now endeavor to hold the backlash of Packard gears within closer limits and work to keep it near to standards, which must be determined for the different designs. Some assemblies will give good results with much more backlash than will others. The high limit is determined invariably by the existence of some rattle; the low limit is determined by the smallest clearance that can be used with oil as thick as that in frequent use in winter weather, without causing growl or bearing noise.

Rear-Axle Bevel-Gears

It has been our experience that whenever you silence one noise in a car, another becomes evident that never caused complaint before. The eventual attainment of silence is the result of a persistent noise-curing campaign which starts with the noise that is most noticeable and works down the line. This method has resulted in finding that the rear-axle noise still persists in most cars though in a much less disagreeable degree.

The spiral bevel-gear, originally introduced by the Packard company, was a big step in the direction of reducing rear-axle gear-noise. Until other units of the automobile had been perfected to their present state of quietness, we were satisfied that rear axles were about as quiet as they could be made commercially. We are now endeavoring to perfect the assembling of the gear so that the noise of operation shall be reduced still more.

If the presentation of these notes results in a valuable discussion of the subjects treated, we shall feel repaid for our efforts in getting the material together.

Stresses and Wear in Automobile Tires[1]

A SURPRISINGLY large proportion of the total energy losses in rubber-tired automobiles has been shown to be due to the rolling-resistance of the wheels and especially of the driving-wheels. How a tire can absorb such relatively large amounts of power has been discussed by Dr. A. Riedler,[2] who concludes that the work done in deforming the tires represents the major part of the tire losses. While Riedler's conception is very plausible, the details of the process by which the work of tire deformation dissipates engine power need further elucidation. It is one of the objects of this paper to present experimental evidence in support of the view that tire losses are due to work expended in tire deformation.

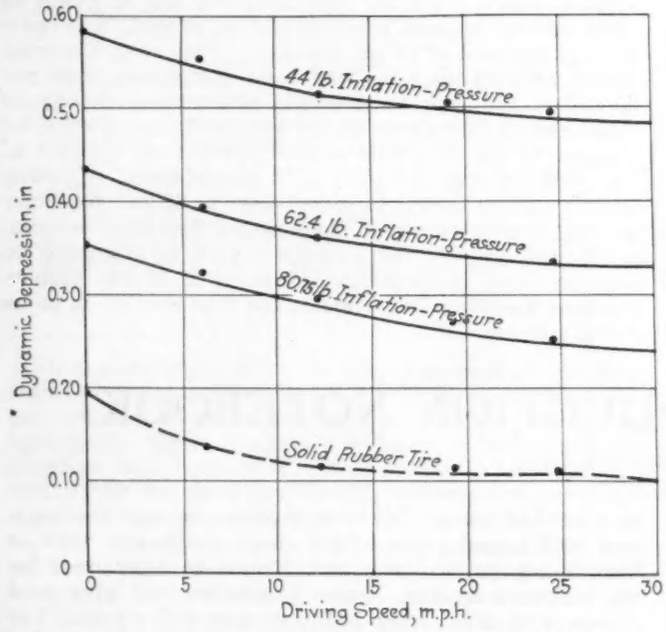


FIG. 1—CHART SHOWING THE VARIATION IN THE DEPRESSION OF TIRES WITH THE DRIVING SPEED AND THE INFLUENCE OF THE INFLATION-PRESSURE ON THE DYNAMIC WHEEL-LIFT

The results of this investigation will be presented in two parts, the first of which relates to conditions on a smooth road, while the second takes into account the effect of obstructions on the road. The experiments discussed in the first part are designed to interpret rolling-resistance in terms of tire deformation as well as to correlate tire stresses and wear. In the second part an attempt is made to extend Riedler's conclusion to the interaction of springs and tires.

STATIC AND DYNAMIC CHARACTERISTICS ON A SMOOTH ROAD

The curve obtained by plotting the deflections or depressions of a rubber-tired wheel at various loads may be called the static characteristic of the tire. A striking fact about such static-characteristic curves is that their trend differs according as the measurements are made at increasing or at decreasing loads. The same peculiarity is shown also by the deflection-load curves of leaf springs where there is friction between the surfaces of the leaves of the spring. In the latter case, the area enclosed between the loop obtained by plotting the deflections corresponding to loads increasing at first up to a maximum and then decreasing back to no load is a direct measure of the work done by the frictional forces between the leaves of the spring. The exactly similar loop shown by the static characteristic of rubber tires suggests at once a similar origin; that is, it may be taken as evidence of frictional sliding between the layers of the rubber tire, and the area of the loop is then a direct measure of the internal frictional forces called into being by tire deformation. In our experiments this internal frictional loss was 5 per cent of the total work of deflection in the case of a solid rubber tire, and from 20 to 30 per cent, according to the inflation, in the case of pneumatic tires.

The deflection or depression determined at varying loads while the wheel is running on a revolving drum was found to be always distinctly smaller than the corresponding static value. Fig. 1 shows how this depression varies with the driving speed in the case of a solid rubber tire and of a pneumatic tire at three inflations, the load remaining constant throughout. These curves show that there is an actual lift of the wheel as the speed increases, this dynamic wheel-lift amounting to 33 per cent of the normal static depression in the case of a pneumatic tire at 80¾ lb. per sq. in. inflation-pressure, and to 53 lb. in the case of the solid rubber tire.

Fig. 1 shows also the influence of the inflation-pressure on the dynamic wheel-lift. A careful comparison of the three upper curves reveals a slight increase of the wheel-lift with the inflation-pressure; for example, the lift for a speed of 19 m.p.h. at 44 lb. per sq. in. pressure is 11 per cent and at 80¾ lb. per sq. in., 14 per cent of the static depression.

This wheel-lift is made more evident by comparing the static characteristic with the dynamic characteristic at some constant speed, as is done in Fig. 2. Here it will be seen that, at a speed of 19 m.p.h., the dynamic wheel-lift is practically independent of the load.

The cause of dynamic wheel-lift may be sought in the action of the inertia forces that resist the sudden accelerations with which the parts of the tire on a revolving wheel are deflected at the moment of contact with the road. The internal frictional forces also are a function of the velocity, for an attempt to slip one layer of tire material suddenly over another will be met by a counter resisting force. Such resisting forces then oppose the sudden deformation of the tire contour, with the result that there is formed an accumulation of tire material immediately in front of the area of contact with the road. This bulge formation, as we shall call it, is compensated for by a decrease in the inertia and friction forces and provides a gradual transition at the point where flattening begins. In effect, this means a smaller deflection of the wheel at a given load.

This view is consistent also with the increase in the wheel-lift with increasing inflation-pressures. The pressure between the layers of tire material increases with the inflation-pressure and, assuming a constant coefficient of friction, the frictional forces will increase as the inflation-pressure increases, and this in turn results in an increased bulge formation and hence a smaller deflection. Load does not affect the lift because the change in the inflation-pressure due to flattening is negligible and, furthermore, the radius of curvature at the point where the tire flattens, a factor that should influence friction, may be regarded as constant in view of the smallness of the depression when compared with the diameter of the wheel.

ROLLING-RESISTANCE

The resultant of all forces acting in a tire represents the rolling-resistance, and can be determined by measuring the rolling losses. This has been done for a solid rubber tire

[1] This paper is a brief adaptation of an article by Otto Enoch published in *Der Motorwagen*, Sept. 10, 20 and 30, 1921. Comment by the Research Department is included, and this is distinguished from the text by enclosing brackets.

[2] See The Scientific Determination of the Merits of Automobiles, p. 314.

and for a pneumatic tire inflated to a pressure of 62.4 lb. per sq. in. The two sets of curves in Fig. 3 show the relation between rolling-resistance, driving speed and wheel load.

An increase in the rolling-resistance with the driving speed might have been predicted as a consequence of the known influence of the velocity on the forces of inertia, but there is another factor contributing to the rate of this increase. This additional loss of power at higher speeds is due to the time lag in the elastic forces that restore the flattened tire to its normal shape; these forces of restitution are damped by friction and by mass-inertia to such an extent that they may be active for some time after the affected part of the tire has left the road, and hence at higher speeds a fraction of the available compression energy is not returned to the tire.

The internal and external frictional forces increase as the flattened area of the tire increases and also as the inflation-pressure increases. The influence of increased surface is shown in Fig. 3 by the greater rolling-resistances at the higher loads, and is confirmed by the two curves taken at a

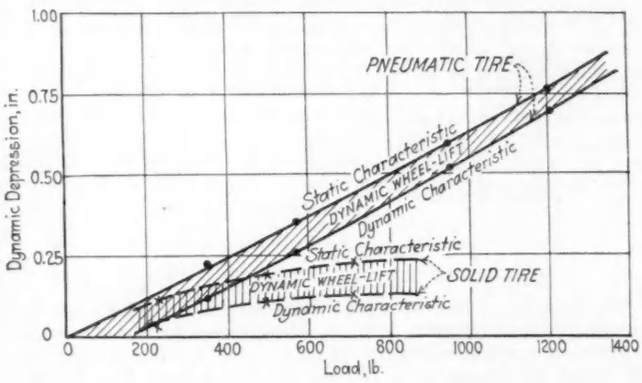


FIG. 2—COMPARISON OF THE STATIC AND DYNAMIC CHARACTERISTICS OF SOLID AND PNEUMATIC TIRES

constant speed of 19 m.p.h., for, since a pneumatic tire flattens much more than a solid tire with increasing loads, the slope of a resistance-load curve should be greater for the pneumatic tire.

A further illustration of the connection between the frictional force and the area of road contact is given in Fig. 4, which shows two sets of curves for the relation between the rolling-resistance and the inflation-pressure. In the set of two curves, the load being constant, there is a smaller area of road contact as the inflation-pressure increases and hence a smaller rolling-resistance, despite the fact that the in-

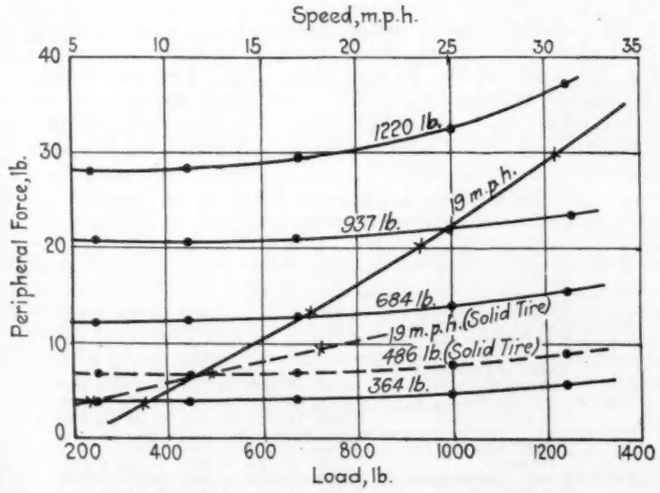


FIG. 3—RELATIONS EXISTING BETWEEN ROLLING-RESISTANCE, DRIVING SPEED AND WHEEL LOAD

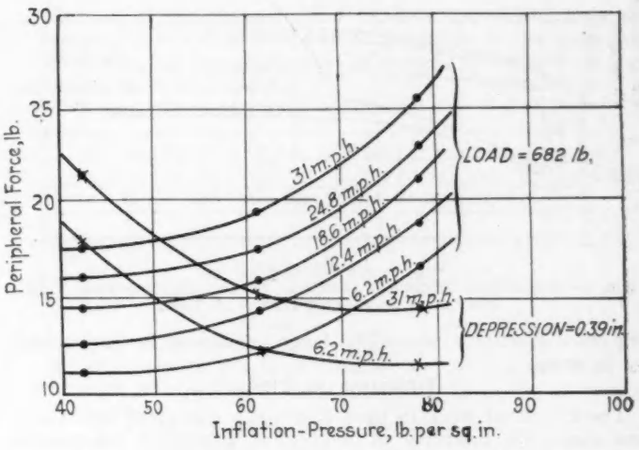


FIG. 4—CHART SHOWING THE RELATION BETWEEN THE ROLLING-RESISTANCE AND THE INFLATION-PRESSURE

creased pressure actually makes the friction somewhat greater. That the internal pressure really has an effect in the sense predicted is proved by the set of five curves in Fig. 4, which were obtained at a constant area of contact; that is, by adjusting the load until the dynamic depression was 0.39 in. for each measurement. In other words, of the two factors determining the magnitude of the frictional work, the area of contact preponderates to such an extent as to mask the pressure effect.

ZONES OF MAXIMUM WEAR

If now the points of the greatest deformation in a tire are the points of the greatest power-absorption and dissipation, one would expect to find the zones of the greatest wear at those points where flattening causes the greatest change in the radius of curvature of a tire section. This would obviously be transversely at the side perpendicular to the driving direction, while longitudinally in the direction of rolling the bending is more gradual.

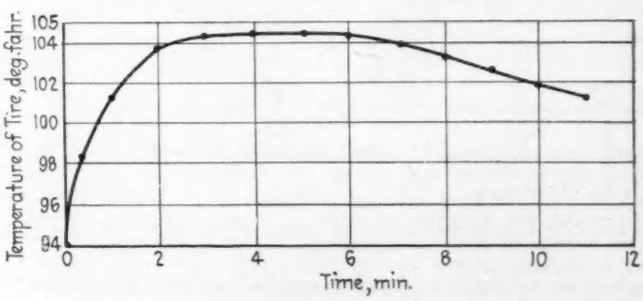


FIG. 5—CURVE SHOWING HOW THE TEMPERATURE OF A TIRE INCREASES AFTER A SUDDEN STOP

Long-run durability tests made on smooth 32½ x 5-in. Calmon tires proved that such is indeed the case. The photographs reproduced in the German original show unmistakably the two lateral zones of maximum wear separated by a zone in which there is a certain amount of accumulated rubber material. Furthermore, the lateral zones are widest in tires of least inflation, as is to be expected according to the view here advanced.

Anti-skid tires, recently appearing on the market in Europe, are very favorably constructed as regards resistance to wear. The ribs on such anti-skid tires present a reinforced surface at the two lateral zones of maximum wear and are, nevertheless, advantageously distributed for the bulge formation. In addition, the ribs should aid materially in the ventilation necessary to keep the temperature of the tires as low as possible.

Wear on the bearing surface of tires is caused also dur-

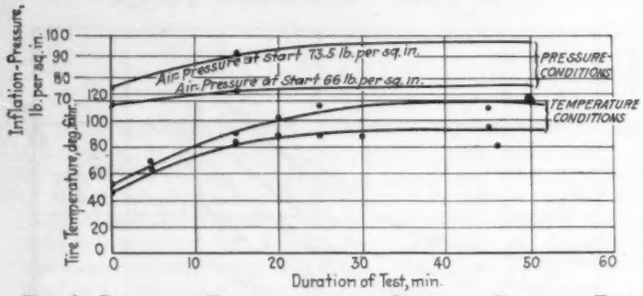

Inflation-Pressure, lb. per sq. in.
Tire Temperature, deg. fahr.
Duration of Test, min.
Air Pressure at Start 13.5 lb. per sq. in.
Air Pressure at Start 66 lb. per sq. in.
PRESSURE-CONDITIONS
TEMPERATURE CONDITIONS

FIG. 6—RELATIONS EXISTING BETWEEN INFLATION-PRESSURE, TIRE TEMPERATURE AND DURATION OF TEST

ing the distortions induced by large variations in torque and by braking.

HEATING OF TIRES

The frictional work in tires is directly converted into heat, and since the quantity of heat thus generated represents from 10 to 20 per cent of the total engine-power, it is not inconsiderable. If the wheel were suddenly stopped, a large part of this heat would go to raise the temperature of the tire, as is shown in Fig. 5.

Under driving conditions, a thermal equilibrium is reached at which the temperature of the tire has risen to such a degree that the gradient toward the surroundings is sufficient to carry off the surplus heat as fast as it is generated.

The higher temperature expands the air in the tire, causing an increase in the inflation-pressure which may burst the tire, for instance on sudden stopping after a fast run. Besides this danger of bursting, the life of the tire is very adversely affected at higher temperatures. Fig. 6 reproduces some of our data on the rise in tire temperature and pressure during driving.

[Details of the experimental arrangement by which these figures were obtained are not given in the text, so that it is

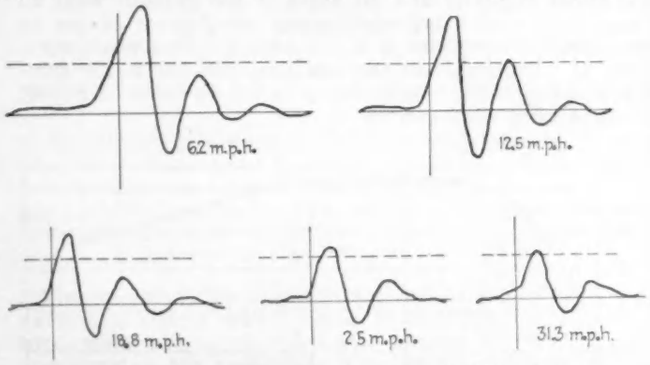


FIG. 7—DIAGRAM SHOWING THE OSCILLATIONS FOLLOWING THE PASSING OVER AN OBSTRUCTION AT VARIOUS SPEEDS

impossible to compare Enoch's results on tire temperatures with those of Lockwood,[3] and of Ellenwood.[4]]

With inferior reclaimed rubber tires we recorded a temperature rise of over 50 deg. fahr.; that is, from 66 to 116 deg. fahr.

The necessity for considering the thermal relations in tire construction is made evident by these results. As already indicated, the ribs of anti-skid tires aid in the reduction of the tire temperature, by the larger radiating surface presented as well as by the additional air-circulation that they induce. Another proposal for a better heat-dissipation is the use of rim material having good thermal conductivity, for example aluminum. Metal buttons sunk into the tire might serve the purpose were it not for possible deleterious effects on the mechanical properties of the tire.

IMPACT STRESSES

In driving over obstructions, tire deformation is of a different nature from that previously described. To get an insight into the effect of a rough road on rolling-resistance, we have recorded the oscillations resulting from vertical impact and calculated approximately the corresponding work of deformation.

Impacts on the wheel were produced by a strip fitted on the revolving drum of the test-stand; the strip was adjusted to strike the wheel once and then drop off. Diagrams of the vertical oscillations of the wheel axle were obtained by using a specially designed device for timing and recording the vibrations. For pneumatic tires, we used a 16-40 hp. Benz chassis and impact strips of 1-in. height, the 32 x 4½-in. tires being loaded to 682 lb. For solid tires, we used a 10-21 hp. Horsch car and a 0.4-in. height of impact strip.

Figs. 7, 8 and 9 and Tables 1 and 2 reproduce some of our data. The oscillation diagrams will be considered as consisting of four phases as follows:

TABLE 1—DIMENSIONS OF OSCILLATION CURVES FOR PNEUMATIC TIRES

Phase 1	Depth of Depression, in.	0.985	1.180	1.230	1.260	1.280
	Length of Period, in.	1.480	0.740	0.490	0.370	0.290
	Car Speed, m.p.h.	6.200	12.500	18.800	25.000	31.300
Phase 2	Height of Throw, in.	0.840	0.760	0.550	0.430	0.380
	Length of Period, in.	0.240	0.490	0.550	0.650	0.700
	Car Speed, m.p.h.	6.200	12.500	18.800	25.000	31.300
Phase 3	Depth of Depression on Return Throw, in.	0.340	0.350	0.246	0.196	0.145
	Length of Period, in.	0.390	0.630	0.590	0.630	0.590
	Car Speed, m.p.h.	6.200	12.500	18.800	25.000	31.300
Phase 4	Height of Second Throw, in.	0.300	0.350	0.220	0.158	0.118
	Length of Period, in.	0.860	0.860	0.860	0.900	0.940
	Car Speed, m.p.h.	6.200	12.500	18.800	25.000	31.300

The first phase starts at the point where the wheel begins to rise from its normal level as determined by the dynamic depression corresponding to the particular driving-speed. In the oscillation diagrams, the horizontal dashed line is the zero from which these dynamic depressions, represented by the horizontal full lines, are measured. The first phase ends at the point where the oscillation curve crosses the vertical line drawn to indicate the middle of the impact strip.

In Tables 1 and 2 the first column is obtained by subtracting the vertical intercepts from the sum of the dynamic depression and height of impact strip; this difference is the amount by which the tire is indented at the moment of rolling over the impact strip. The second column gives the horizontal distance between the initial point of the curve and the impact strip. The second phase includes the oscillation from the middle of the impact strip to the point where the wheel again reaches its normal position for a smooth road. The third and fourth columns of the tables give respectively the vertical throw and the horizontal distance traveled during the second phase, the former measured from the point of contact of the wheel with the middle of the impact strip to the highest point of oscillation curve, and the latter corrected for the width of the impact strip. The third and fourth phases are represented by the two succeeding loops of the curve. The small oscillations after the fourth phase have been found to have no influence on the work of deformation.

TABLE 2—OSCILLATION CURVES FOR SOLID TIRES

Phase 1	Depth of Depression, in.	0.287	0.390	0.430	0.470	0.490
	Length of Period, in.	0.900	0.450	0.290	0.230	0.190
	Car Speed, m.p.h.	6.250	12.500	18.700	25.000	31.200
Phase 2	Height of Throw, in.	0.630	0.770	0.820	0.630	0.630
	Length of Period, in.	0.250	0.730	0.860	0.720	0.840
	Car Speed, m.p.h.	6.250	12.500	18.700	25.000	31.200
Phase 3	Depth of Depression on Return Throw, in.	0.180	0.320	0.310	0.290	0.290
	Length of Period, in.	0.270	0.350	0.310	0.350	0.350
	Car Speed, m.p.h.	6.250	12.500	18.700	25.000	31.200
Phase 4	Height of Second Throw, in.	0.150	0.200	0.310	0.190	0.220
	Length of Period, in.	0.470	0.550	0.630	0.510	0.550
	Car Speed, m.p.h.	6.250	12.500	18.700	25.000	31.200

[3] See TRANSACTIONS, vol. 12, part 1, p. 380.

[4] See THE JOURNAL, August, 1922, p. 129.

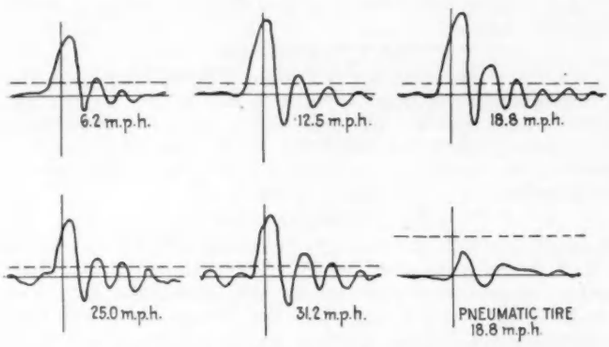


FIG. 8—ANOTHER OSCILLATION DIAGRAM

An examination of Figs. 7 and 8 shows that as the driving speed increases, both the depth of original depression and the amount of throw increase. Furthermore, the horizontal distances traveled during the half-oscillations below the normal position are smaller than the distance traveled during the half-oscillations above the normal position; this is particularly noticeable in the third and fourth phases. This inequality in the vibration periods is caused by the higher equilibrium position of the axle as it is raised by the upward swing of the frame of the car; the frame begins its upward swing at the end of the second phase and, because of the slower oscillation of the heavy frame, continues over phases three and four of the wheel oscillation.

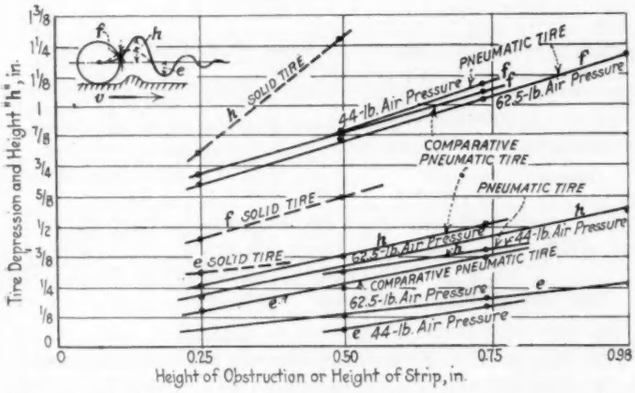


FIG. 9—CHART SHOWING THE RELATION BETWEEN THE HEIGHT OF THE OBSTRUCTION OR THE IMPACT STRIP AND THE RESULTING TIRE DEPRESSION

A comparison of pneumatic and solid rubber tires as regards behavior during impact reveals the much larger amplitude of oscillation in the case of the solid tire. In the diagrams this is shown by a greater height above the horizontal dashed line for the first throw, by a greater depth below the horizontal full line of the return throw, and by the number of half-oscillations that rise above the dotted zero-line. To illustrate these differences, Fig. 8 includes for comparison an oscillation curve obtained after substituting pneumatic tires for the solid tires on the Horsch car. In Fig. 9 this larger throw is shown by the fact that the h-curves for the throw are below the f-curves for the depression in the case of pneumatic tires, but above the f-curves for solid tires. In Fig. 9 the height of the impact strip is plotted against the throw and the wheel depression for two inflation-pressures of pneumatic tires on the Benz chassis and for solid tires together with a comparison set of pneumatic tires on the Horsch car. Lines marked h refer to upward throw, e to return throw, and f to depth of first impact-depression.

ROLLING-RESISTANCE DURING IMPACT

The work of tire deformation during impact may be calculated approximately from the diagrams of the previous section, in connection with the relation between rolling-resistance at various loads and dynamic depression as measured in connection with the work described in the first part.

We have for any depression of e feet at a speed of v ft. per sec.,

$$d\,W = 550\,L\,dt$$

where

dW = the work done, in foot-pounds and

L = the horsepower per second lost in producing dynamic depression at any instant

If for dt we substitute $\frac{dx}{v}$, where x is the horizontal distance in feet on our oscillation diagram, we have

$$W = \frac{550}{v}\int_{x_2}^{x_1} L\,dx \qquad (1)$$

To evaluate L, we have plotted the results given in the first part of the paper for the power losses L, at various dynamic depressions e. The curves thus obtained approximate very closely to straight lines and therefore we may write

$$L = ce \pm a$$

c and a being constants. Equation (1) then becomes

$$W = \frac{550c}{v}\int_{x_1}^{x_2} e\,dx \pm \frac{550a}{v}\,(x_2 - x_1) \qquad (2)$$

The definite integral $\int_{x_1}^{x_2} e\,dx$ is the area included between the curve and the horizontal axis, and may be evaluated planimetrically for each one of the four phases of the oscillation diagrams. It happens that the straight lines in the L e diagram for pneumatic tires go through the origin and hence the second term of equation (2) has a value for solid rubber tires only. Table 3 gives the values calculated for c and a, where c is the horsepower per inch of dynamic depression and a is the horsepower; and Fig. 10 shows the

TABLE 3—VALUES OF CONSTANTS c AND a

Car Speed, M.P.H.	Type of Tire		
	Pneumatic, c	Solid Rubber	
	Inflation-Pressure, 62.5 Lb. per Sq. In.	c	a
6.25	0.556	0.739	0.00
12.50	1.280	1.817	0.00
18.70	2.145	2.812	0.06
25.00	3.107	4.082	0.15
31.20	4.224	5.384	0.31

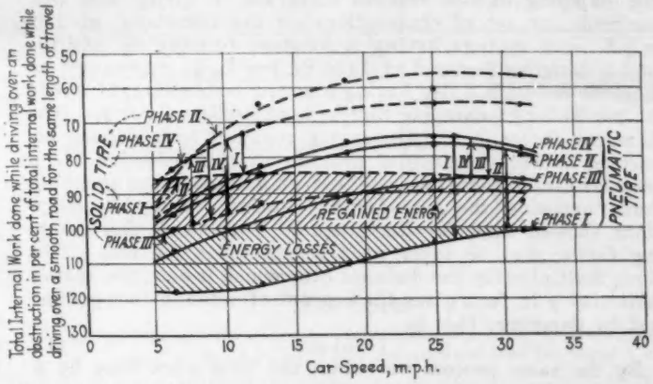

Total Internal Work done while driving over an obstruction in per cent of total internal work done while driving over a smooth road for the same length of travel
Car Speed, m.p.h.
SOLID TIRE
PNEUMATIC TIRE
PHASE I
PHASE II
PHASE III
PHASE IV
REGAINED ENERGY
ENERGY LOSSES

FIG. 10—CHART SHOWING THE DISSIPATION OF ENERGY IN THE TIRE WHEN DRIVING OVER AN OBSTRUCTION

values of W in equation (2) for equal horizontal distances of travel.

It will be noticed that the first and second phases practically determine the rolling-resistance, and that the third and fourth phases so nearly counterbalance as to be equivalent to a correction-term only. The second phase may therefore be said to yield a positive gain in elastic work which nearly equals or exceeds the large loss in power occasioned by the high surface-pressure suffered by the tire in the first phase of the impact.

The remarkable conclusion of this calculation is, then, that the work of deformation during impact is not markedly different from the rolling loss on a smooth road, and that indeed the losses on a rough road may actually be less than on a smooth road in the case of stiffer tires, pneumatic with high inflation-pressures or solid rubber, at high driving-speeds.

[We believe that this conclusion cannot be accepted because the principle underlying the calculations on which the conclusion is based is open to grave objections. In the first place, it is not obvious that the power loss at a given wheel deflection during an impact oscillation is the same as the power loss determined at the same depression when running on a smooth road; in fact, this assumption involves the very point that it is desired to test. Furthermore, according to Enoch's contention in the first part of the paper, the friction between the layers of the tire increases with the velocity of depression and thus the power lost on the downward loops of the oscillation curve is more than Enoch calculates, and the power gained on the upward loops, is less.

Even less justifiable is the assumption that when the wheel leaves the road, there is a negative depression of the same type and with the same horsepower factor c as when the wheel is running under load with a consequent positive depression; that is, some of the power-gain obtained planimetrically is certainly fictitious.

Finally, the calculation does not consider several features characteristic of impact, such as the horizontal component of the impact and the spin of the free wheel at the top of its swing. These ultimately result in a power loss, as is shown later in the paper.]

This last conclusion agrees with driving experience according to which no change in the throttle-opening is required to maintain a definite high speed in going from a smooth stretch of road to one of greater unevenness.

[The reference to driving experience is unfortunate at least, because, for the most part, statements such as this are based on very inadequate evidence.]

Interaction of Tires and Springs

Among the factors contributing to tire wear that have not been considered in the above discussion is the frictional work lost in the tire while it is performing the function of a spring. For, during the impact oscillations there is continuous reciprocal interchange of kinetic energy between the springs and the tires; the tire dissipates energy in frictional damping forces just as a spring does.

To estimate roughly the relative losses incurred during the damping of the vertical vibration of spring and tire we took one set of observations on the test-stand, at 18.2 m.p.h. with springs having a constant b, 0.018 in. per lb. and a damping factor k of 0.130 in. per lb. as determined by Bobeth[8] and with a tire having a spring constant, b, of 0.0031 in. per lb. and a damping factor, k, of 0.0350 in. per lb. The damping factor k, like the spring constant b, is defined according to the Continental practice as the deflection for a unit load, and hence the quotient of k/b gives the damping force opposing the force of the spring. Our determinations showed that, for the range of loads used, this damping factor may be taken as a constant for the tire. This force multiplied by the distance over which it acts, the spring deflection y in feet, gives the amount of work in foot-pounds lost by damping; that is,

$$(k/b)\, y \qquad (1)$$

By the same process we obtain the total work done by a spring during a deflection of y feet from an original static depression d as

$$[(d + \tfrac{1}{2}\, y) \div b]\, y \qquad (2)$$

because, since the force varies from d/b to $(d+y)/b$ during the deflection, the average force is $\frac{1}{2}[d/b + (d+y)/b]$, and the work done equals force times deflection. The percentage loss in work is therefore 100 times the work lost by damping, according to equation (1), divided by the total work done by spring deflection, according to equation (2)

$$[k/(d + \tfrac{1}{2}\, y)] \times 100 \qquad (3)$$

Substitution in equations (1), (2) and (3) of the values for b, d, k and y, as obtained in our experiment shows that, despite the smaller deflections of the wheel, the tire does more work in spring-action than the spring itself, but that the damping loss, $0.0350/0.0031 = 11$ lb. per in. of deflection of the tire, is somewhat less than for the spring, $0.1300/0.0108 = 12$ lb. per in. of deflection.

[This deduction must be accepted with reservation, for data on which it is based are not given, nor is the experiment further described. From the context it seems that, after determining b, d, and k for the wheel and the spring, simultaneous oscillation curves for both the wheel and the car body were obtained at five speeds of the impact drum, and that these curves were used to get the y values for the spring and the wheel. The substitution of these y values, probably the maximum upward throw shown by the oscillation curves, in equation (2) gave columns Nos. 2 and 5 of Table 4; in equation (1), columns Nos. 3 and 6; in equation (3), columns Nos. 4 and 7.]

Table 4 gives the results of our experiment on the relative damping-losses in tire and springs.

TABLE 4—CALCULATED WORK DONE BY SPRING AND TIRE DEFLECTION

	Driving Speed, m.p.h.	6.20	12.40	18.60	24.80	31.00
Springs	Work, ft-lb.	40.60	42.90	32.80	24.40	22.20
	Loss, ft-lb.	2.46	2.46	1.82	1.38	1.23
	Loss, per cent	6.00	6.00	6.00	6.00	6.00
Tires	Work, ft-lb.	41.90	52.50	45.40	24.80	30.10
	Loss, ft-lb.	1.31	2.03	1.60	1.30	0.94
	Loss, per cent	4.30	3.90	3.50	3.30	3.10
Sum of Deflections, in.	Springs	1.06	1.10	0.83	0.63	0.55
	Tires	0.87	0.95	0.75	0.63	0.47

The wear in the tire resulting from the friction represented in the damping vibration is of course an important amount to be added to the wear from the impact losses as given in the previous section of this paper.

Our investigations on the impast stresses were performed on automobiles in which provision was made for vertical spring-action only.

[In the German original there is a consideration of resonance effects in the oscillations. Calculation showed that the range of critical speeds was far removed from the possible velocities of the author's experiments.]

Since the impact stresses were found to be dependent almost wholly on the first two phases of the impact, it is obvious that impact stresses in tires, and consequently tire wear, could be diminished by moderating the first two phases of an impact. Theoretically, this could be accomplished by providing springs for the horizontal components of the impact, and our measurements on a car, fitted with an Arop pendulum spring, confirm this theoretical deduction.

[The author treats spring action at length, but his treatment is omitted here as being only remotely related to tire stresses and wear.]

Figs. 11 and 12 illustrate the favorable action of an Arop horizontal spring.

The import of this improvement in the time and deflection relations of the wheel, as regards the durability of pneumatic tires, may be seen from the following considerations. First, there is the smaller acceleration involved in the inertia forces. The horizontal spring allows the axle to deflect backward, and this is equivalent to a corresponding

[8] See Power Losses and Spring Action in Automobiles, by E. Bobeth, in German.

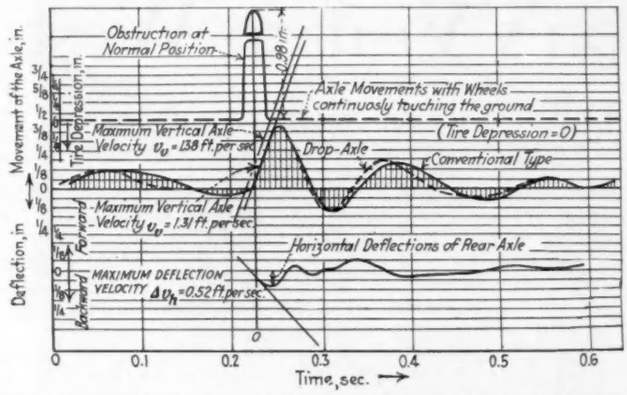

Obstruction at Normal Position
0.98 in.
Axle Movements with Wheels continuosly touching the ground
(Tire Depression = 0)
Maximum Vertical Axle Velocity v_0 = 1.38 ft. per sec.
Drop-Axle
Conventional Type
Maximum Vertical Axle Velocity v_0 = 1.31 ft. per sec.
Horizontal Deflections of Rear Axle
MAXIMUM DEFLECTION VELOCITY Δv_h = 0.52 ft. per sec.
Movement of the Axle, in.
Tire Depression, in.
Deflection, in
Time, sec.

FIG. 11—COMPARISON OF THE OSCILLATION OF A CONVENTIONAL AND AN AROP REAR-AXLE IN PASSING OVER AN OBSTRUCTION 0.98 IN. HIGH AT A SPEED OF 18.7 M.P.H.

lengthening of the period of impact. This softening of the impact blow is equivalent to the action of a tire at a low inflation-pressure, without, however, the disadvantages of the low-pressure tire. Incidentally, but by no means unimportant, is the decrease in the tire distortion resulting from this freedom of the wheels in a horizontal direction.

Second, horizontal springs act as as an elastic coupling between the engine and the road, which prevents the distortions in the tire that would otherwise result from changes in torque, sudden throttling or braking. For example, the sudden release of the tractive resistance when a driving-wheel bounces off the road, speeds up the wheel so that when it returns to the road again there is excessive wear for the short time required for the wheel to resume its normal speed. The following calculation will give some idea of the energy involved in tire distortion of this nature: the driving torque of a 30-hp. passenger-car with 31.5-in. tire-diameter at 31 m.p.h. is about 433 ft-lb. This whole torque is used to accelerate the wheel while it is off the ground. If the weight of the wheel is taken as 66 lb., we have for the polar moment of inertia about the axle

$$m r^2 = J$$

$$J = (66/32) \times [31.5/(2 \times 12)]^2 = 3.56$$

and the angular acceleraton imparted to the wheels by a torque of 433 ft-lb. would be

$$dw/dt = 433/3.56 = 122 \text{ radians per sec. per sec.}$$

If, now, we assume an average duration of bounce of 0.05 sec. during which acceleration is to be constant, we have as the increment in the angular velocity

$$\Delta w = 122 \times 0.05 = 6.1 \text{ radians per sec.}$$

Since a speed of 31 m.p.h. corresponds to an angular velocity of 34.6 radians per sec., we have as the increase in kinetic energy

$$\Delta E = J w \Delta w = 3.56 \times 34.6 \times 6.1 = 750 \text{ ft-lb.}$$

This energy that would ordinarily be transformed into tire distortion and thus into wear, is readily absorbed by a horizontal spring.

In Fig. 12 there are plotted the horizontal deflections h, the height of throw w and the maximum vertical velocity V_v. More detailed analysis of the experimental figures obtained with the Arop spring has led us to the design of other spring-systems which are even more efficient, and it may be emphasized that along this line there is great opportunity for improving the spring-suspension of automobiles and for diminishing the wear on tires.

SUMMARY

The wear of an automobile tire is caused by deformations in the inner layers of the tire and by sliding on the road. These two sources of friction develop heat which increases the temperature of the tire. A decrease in the number of layers in building up the tire and a decrease in the coefficient of friction would therefore increase the durability of the tire.

Wear is concentrated in two lateral zones of the tire. Properly shaped ribs are advantageous in reinforcing a tire along these two zones of wear and, in addition, provide increased ventilation for reducing the temperature of the tire. Rims and wheels made of material possessing high conductivity for heat are also advantageous in the latter respect.

Tires are subject also to impact stresses on rough roads. The work of deformation due to these impact stresses is approximately the same as on a smooth road, but decreases with the inflation-pressure. The first two phases of impact determine the deformation work.

Impact stresses can be reduced by decreasing the inertia-mass; that is, by decreasing the weight of wheels, axles and other unsprung parts; also by decreasing the impact accelerations. In this connection, tires with lower inflation-pressure combine the advantage of lessening the impact with the disadvantage of greater deformation work. Horizontal springs possess both an effective impact-damping and a decelerating action of the inertia-masses.

Horizontal spring systems, in addition to other desirable qualities, contribute to an increased durability of the tire.

AMERICAN AND EUROPEAN PRACTICE

[In reference to cord and fabric tires here and abroad, the original series of articles published in *Der Motorwagen*[6] of which the foregoing is a critical digest, contained a statement, which has appeared in other European journals also, to the effect that cord tires used in Germany show a consistently lower mileage and greater power loss than fabric tires.

This statement was so much at variance with both laboratory and service experience in this Country that a letter was written to the author of the articles, Dr. Otto Enoch, asking for further particulars. A translation of our letter appears in a later issue of *Der Motorwagen*,[7] together with a translation of the paper by A. B. Browne and Professor Lockwood entitled Practical Testing of Motor Vehicles[8] and a statement that the letter would be answered later by Professor Enoch.

The very wide discrepancy between American and European experience is so interesting that we should like to hear from any of our members who can throw further light on the subject.]

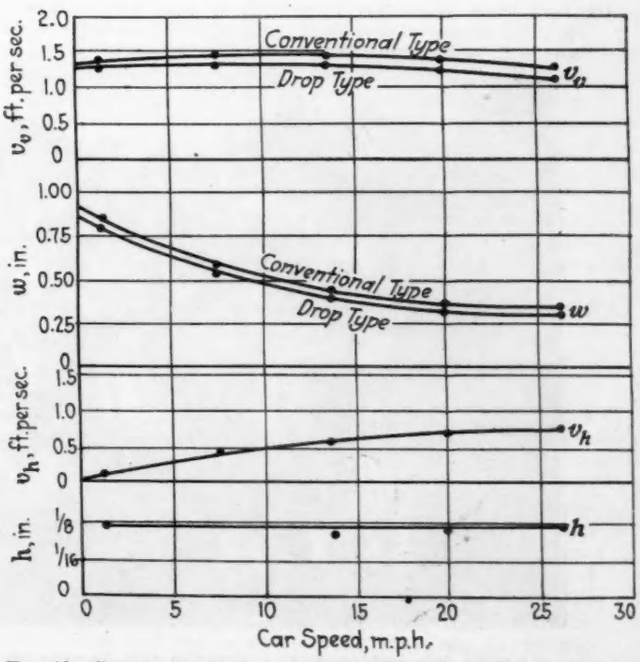


FIG. 12—CHARACTERISTICS OF THE AROP REAR-AXLE SPRING-SUSPENSION IN PASSING OVER AN OBSTRUCTION 0.98 IN. HIGH

[6] See *Der Motorwagen*, Sept. 10, 20 and 30, 1921.

[7] See *Der Motorwagen*, Aug. 10, 1922, p. 413.

[8] See TRANSACTIONS, vol. 10, part 1, p. 68.

Metropolitan Section's Visit to West Point

OF all the summer outings held by the Metropolitan Section, the one of Sept. 16 last will undoubtedly stand high in the memory of those who were fortunate enough to be there. The lazy trip up and down the Hudson River was restful and gave plenty of opportunity to make new friends and talk business or pleasure at the whim of the moment. But the crowning feature of the trip was the visit to the United States Military Academy.

We arrived at the West Point dock at 2:15, almost an hour behind schedule, and were informed that by making a quick trip up the hill we could see the end of the daily inspection. So the crowd immediately boarded the various forms of conveyance, buses, taxicabs, etc., and gained the top in ample time to see the corps of cadets, 1300 strong, drawn up for inspection. In their gray full-dress uniforms, with white crossed belts and high hats, the cadets made a wonderful picture against the green parade-ground, with its background of trees and distant mountains. The Academy Band, in its black and white full-dress, played during the inspection and as the cadets marched off to their barracks.

Immediately afterward we were divided up into sections, and an officer was on hand to act as guide for each section. The sections were headed by Major Charles G. Mettler, professor of ordnance and gunnery, Majors Oscar J. Gatchell and Robert N. Bodine and Capt. Rudolph F. Whitelegg. All these officers belong to the Ordnance Department and are detailed to the Department of Ordnance and Gunnery at the Academy. Under their able and sympathetic guidance the various sections were taken to see where the cadets ate, slept, lived and played.

At the cadets' gymnasium, the fencing, wrestling and boxing rooms, the large gymnasium for floor drill, the swimming pool, the indoor rifle and pistol range were all exhibited and explained. Here, under the direction of the Master of the Sword, Lieut.-Col. H. J. Koehler, the new cadets with bodies partly developed and poorly carried are given an intensive course of physical training. The exercises round out their bodies, develop a correct posture and improve their voices for command, so that within 6 weeks they appear on parade marching erect and steady like the older cadets. The training and athletics are intended not only to develop the individual but also to help him in his future work of training soldiers of the American Army. Each cadet is taught to fence with foil and sabre and to box, wrestle and swim. Outside, he is taught golf, tennis, lacrosse, soccer, baseball, and football.

One of the rooms where the cadets sleep and study was next visited. The room that we saw in the North Barracks was used by three cadets, although built only for two. But on account of the congested condition of the barracks, and the failure of Congress to appropriate funds for extension, the room was fitted with a double-deck cot where two men slept, with the third using a single cot. Our attention was called to the arrangement of books and clothing, as this was provided for by the rules and was alike in every cadet's room.

Each of the 12 companies in the corps has a room set aside for the tactical commander, who supervises the discipline and supplies of the organization. In this room, one of which we inspected, the officer in charge is accessible to cadets for advice and suggestion, as well as for the carrying out of discipline.

The next stop was at the beautiful Memorial Hall, the design of Stanford White and the gift of General Cullum. Here are recorded on bronze tablets the deeds of graduates who have been killed on various battle-fields. Set into the walls are many trophies of old wars, such as cannons and arms. The second floor of the Memorial Hall is used for cadet hops, lectures and dramatic performances, although it is entirely inadequate in size. When dances are given, for example, the room is only large enough to accommodate about one-third of the corps.

The balcony of the Memorial Hall, 130 ft. above the river, presents a wonderful view of the great water-gap of the Hudson, which lies between Storm King and Breakneck mountains. On the day of our visit the blue sky was dotted with white clouds so that the mountains were outlined sharply. Underneath was the Hudson, deep blue as the shadows fell across it in the late afternoon. It was easy to

MEMBERS AND GUESTS OF THE METROPOLITAN SECTION AT WEST POINT, SEPT. 16, 1922, ON THE ANNUAL OUTING OF THE SECTION

VIEW OF THE UNITED STATES MILITARY ACADEMY AT WEST POINT FROM AN AIRPLANE

believe the statement of our officer guide, that here is one the most beautiful scenes in America.

The view also indicated why in Revolutionary days West Point became the stronghold of the Hudson. At first the east point of the river on Constitution Island was fortified, but when General Washington, accompanied by Kosciusko, Lafayette, Von Steuben and others of his immortal staff, surveyed the situation, they immediately directed the fortification of the west point of the river, and the name has since been retained. On the hilltops above the academy lies Fort Putnam and other redoubts. These were pointed out and their use during the Revolutionary War explained to us. Also exhibited were a few links of the old chain that was stretched across the river to prevent the passage of the British Fleet during the Revolutionary War.

There is not space here to tell about all the things we saw, but mention should be made of the gilded monument presented to the Academy by a sister school, the Ecole Polytechnique of France, in commemoration of the entrance of the United States into the World War. Beyond Trophy Point, a short promontory used to display cannons and relics of old wars, we visited the old Ordnance Laboratory, built in the early '30's of the last century as a central fort to protect ammunition and supplies.

At the Laboratory were gathered the different types of motor-transport vehicle used on the reservation. These included machine-shop and supply trucks, ordnance tractors, staff cars and motorcycles. Motor-transport officers at the Point were on hand to exhibit the vehicles and explain their operation to the members of our party.

A few brief speeches were made immediately after the dinner. The Outing Committee, A. C. Bergmann, introduced Chairman Kemp of the Metropolitan Section, who in turn called upon President B. B. Bachman of the Society, who was a guest of the Section for the trip. Mr. Bachman complimented the officers on the way in which the visit had been handled and for the pleasure experienced by all the members of the party. In a few graceful words of acknowledgment, Major Mettler expressed the hope that we would all come again, and also spoke of the need for larger facilities at the Point, in view of the reduction of our Military forces that made more than ever necessary the continuous training of young men who could be competent to turn citizens into soldiers in case of a National emergency.

Those who made the trip with the Metropolitan Section now appreciate, more than ever before, the important part that West Point has played in our National life. Since its establishment, about 6900 cadets have been graduated. These men have fought in all the wars of the United States, have been successful in many prominent positions in civil and political life, and have occupied the Presidential Chair with credit.

The present corps of nearly 1300 cadets comes from all parts of the Country, since each State and every Congressional District have a fixed number to appoint. There are also 180 men from the enlisted force of the regular army and national guard. Upon graduation a cadet is commissioned as second lieutenant in the regular army, where he is expected to serve for at least 4 years.

These cadets are paid $1,174.20 a year, and from this amount provide themselves with clothing and food and the small necessaries of life. Their funds are kept by a treasurer and are not given to the cadets except in such quantities as are required for any journeys away from the Academy that they are permitted to make.—R. E. Plimpton, secretary of Metropolitan Section.

CENTRIFUGAL CASTINGS

COMMENTING editorially upon a brief paper by L. Cammen on Chromium-Alloy Steel Cast Centrifugally,[1] *The Iron Age* says that this probably is the first account of the casting in a centrifugal machine of a high-carbon chromium steel and the successful production of an alloy-steel casting by the centrifugal process. The chief feature is the attainment directly of a structure that only the most careful and time-consuming forging or rolling, with subsequent heat-treatment, can develop in the same kind of alloy-steel cast as an ingot. The centrifugally cast chromium-steel, unworked and heat-treated, is at least equal in structure and properties to similar metal treated by present methods. To this is added the possibility of producing an alloy with higher carbon and chromium-contents than can be secured in ordinary working by the methods employed at present in foundries producing steel castings.

[1] See *The Iron Age*, Sept. 14, 1922, p. 655.

Questions Answered by the Research Department

MANY of the inquiries answered by the Research Department are of much general interest and involve a considerable amount of thought and research. For the information of our members a few of them will be published each month. It is hoped that in this way those members who have not yet availed themselves of the facilities offered by the Research Department may be encouraged to take advantage of them in the future.

Question:—Could you give me the approximate figure for the Brinell hardness of cast iron?

Answer:—Brinell hardness numbers for cast iron of the composition range used for automobile engine cylinders can be found in the *Foundry Trade Journal*, Vol. 21, pp. 811 and 812, where there is given an abstract of results published by the prominent French metallurgist, Leon Guillet; also in an article by J. E. Hurst on The Heat Treatment of Gray Cast Iron at Low Temperatures, in *Engineering*, Vol. 108, p. 1.

The values given in these two articles were obtained with a 10-mm. ball at 3000-kg. pressure, and vary from 179 to 228 for castings cooled in sand. Both authors are interested in the effect on hardness and wear by the use of the castings under conditions where comparatively high temperatures prevail. In fact, the latter paper gives micrographs of an automobile engine piston that failed because of such a temperature influence, while Guillet's curves show that the hardness begins to diminish at about 600 deg. cent. (1470 deg. fahr.).

There are a few other published figures, but as the two independent sets presented above not only agree well, but also relate to conditions obtaining in automobile engine cylinders, they will probably serve the purpose. References:

L. Guillet. Hardness of Cast Iron. *Foundry Trade Journal*, Vol. 21, pp. 811-812.

J. E. Hurst. Heat Treatment of Cast Iron (Gray) at Low Temperatures. *Engineering*, Vol. 108, p. 1.

H. Hatfield. Cast Iron in the Light of Recent Research. 1918.

Question:—Can you supply me with a list of important recent articles and books on (a) bearing friction and (b) determination of viscosities?

Answer:—The following are a few of the recent articles on bearing friction and determination of viscosities:

The Ball Bearing: In the Making, Under Test, and in Service. By Henry Heathcote. Pamphlet can be secured from the Institution of Automobile Engineers, 28 Victoria Street, London, England. Also published in the *Proceedings* for 1920-1921, p. 569.

Friction and Carrying Capacity of Ball and Roller Bearings. By H. L. Whittemore and S. N. Petrenko. Bureau of Standards Technologic Paper No. 201. Published by the Government Printing Office, City of Washington. Gives an account of experiments undertaken to determine maximum safe load and static friction under load of ball and roller bearings. Results agree roughly with Hertz's theory, differences being ascribable to inhomogeneity of material.

Viscosity and Friction. By Winslow H. Herschel. THE JOURNAL, January, 1922, p. 34.

Macmichael Torsional Viscosimeter. By W. H. Herschel and E. W. Dean. Bureau of Mines Serial No. 2201, January, 1921.

Saybolt Furol Viscosimeter. By E. W. Dean. Bureau of Mines Serial No. 2215, February, 1921.

Determination of Absolute Viscosity by Short-Tube Viscosimeters. By Winslow H. Herschel. Bureau of Standards Technologic Paper No. 100. Published by the Government Printing Office, City of Washington.

Question:—The variation in the performance of trucks introduces some difficulties in interpreting results. We understand that the best passenger cars are designed with a view to securing an acceleration of about 3 ft. per sec. per sec. Is a similar basis being adopted for the design of motor trucks in terms of acceleration or power per ton of weight? Can you give us an idea of the range of tractive effort of various sizes of truck?

Answer:—Using the best estimates we could make for engine torque and mechanical efficiency, and 35 lb. per ton for tractive effort on average roads, the maximum grade and accelerations in direct or high gear work out as shown in the accompanying table. The trucks were selected very much at random, but represent for the most part models that, we believe, are put out in considerable numbers. The weights used in the following table are the chassis and body weights as given by the makers, plus the *rated* load. The figures can be readily recomputed for any desired load. A brief statement of the methods used in computing these results is given below. For obvious reasons, the names of the trucks are not given.

Truck	Rated Capacity, tons	Bore, in.	Stroke, in.	Gear-Reduction	Diameter of Rear Wheel, in.	Total Weight, lb.	Grade, per cent	Mean of Groups[1]
1	½	3 7/8	4 ½	4.16	32	3,850	6.04	6.16 per cent grade.
2	¾	3 1/8	4	5.50	34	4,119	5.54	1.98 ft. per sec. per sec.
3	¾	3 ½	5	6.30	34	4,500	6.90	
4	1	3 ¾	5	6.28	34	5,300	6.65	7.11 per cent grade.
5	1	3 ¾	...	6.20	34	5,300	6.53	2.29 ft. per sec. per sec.
6	1	3 ¾	4	7.25	32	4,400	8.15	
7	1½	4	5 ¾	7.00	36	7,000	6.93	6.05 per cent grade.
8	1½	3 ¾	5	7.60	36	7,000	5.52	1.95 ft. per sec. per sec.
9	1½	3 ¾	5	7.80	34	7,400	5.69	
10	2	4 ⅛	5 1/12	8.00	40	8,800	5.00	4.85 per cent grade.
11	2	4	5	7.60	36	8,800	4.83	1.56 ft. per sec. per sec.
12	2	4	5 ½	6.43	34	8,800	4.73	
13	2½	4 ⅛	5 ¼	9.00	38	10,650	5.00	4.12 per cent grade.
14	2½	4 ⅛	5 ¼	7.75	36	11,000	4.25	1.33 ft. per sec. per sec.
15	2½	3 ¾	5 ¼	8.66	36	11,000	3.79	
16	2½	4	5	7.99	36	11,700	3.43	
17	3	4 ¼	5	9.00	36	11,700	4.84	4.57 per cent grade.
18	3	4 ⅛	5 ¼	9.45	36	13,200	4.39	1.47 ft. per sec. per sec.
19	3	4 ¼	5 ½	9.66	36	14,500	4.52	
20	3½	4 ½	5 ½	10.26	36	15,400	5.30	4.97 per cent grade.
21	3½	4 ½	6	8.75	40	15,400	4.17	1.60 ft. per sec. per sec.
22	3½	4 ½	5 ½	10.33	36	15,400	5.36	
23	3½	4 ½	6 ½	10.00	40	16,500	5.06	
24	4	4 ½	5 ½	10.25	36	17,600	4.42	3.79 per cent grade.
25	4	4 ½	6	8.75	40	18,500	3.15	1.22 ft. per sec. per sec.
26	5	5	6 ¼	8.80	40	22,000	3.60	
27	5	4 ½	6 ¾	10.00	40	21,800	3.60	3.99 per cent grade.
28	5	5	5 ½	10.66	40	23,150	3.65	1.29 ft. per sec. per sec.
29	5	4 ¼	5 ½	8.70	36	22,000	1.99	
30	5	5	6 ½	11.66	40	23,500	5.11	4.58 per cent grade.
31	7½	5	6	11.58	36	26,050	4.58	1.48 ft. per sec. per sec.

[1] Second decimal has no significance.

The method used in the calculation of the tractive effort is given by Cornelius T. Myers in an article entitled Power and Performance of Gasoline Motor Trucks[2] and also in one entitled Chassis Design of Class B Motor Trucks[3]. Accord-

[2] See TRANSACTIONS, vol. 9, part 2, p. 122.

[3] See THE JOURNAL, January, 1922, p. 29.

ing to the former the tractive factor is the tractive effort in pounds divided by the total weight in pounds on the tires or the product of the engine torque and the gear reduction divided by the product of the total weight in pounds on the tires and one-half of the diameter of the driving wheel. The torque is taken as 1.2 of the value corresponding to the National Automobile Chamber of Commerce rating.

$$T = [\{ (33{,}000D^2n \times 12) \div 2.5 \} \div \{ (2\pi \times 1000 \times 12) \div 2L \}] \times 1.2$$
$$= 4.2D^2nL \times 1.2$$
$$\therefore TF = [(8.4D^2nLe) \div Wd] \times 1.2e_m$$

where

d = the diameter of the rear wheel
D = the cylinder bore
e = gear reduction
e_m = the mechanical efficiency
L = the length of stroke
n = the number of cylinders
T = the torque
TF = the tractive factor
W = the total weight on the tires

Assuming a mechanical efficiency of 85 per cent, the formula for four-cylinder engines becomes

$$TF = [(8.4 \times 1.2 \times 0.85 \times 4)D^2Le] \div Wd$$
$$= 34.27D^2Le \div Wd$$

This gives the total tractive effort for each pound of weight of the truck. By subtracting the tractive resistance per pound on good dirt roads, which is taken as 35 lb. per ton or 0.0175 lb. per lb., the tractive effort that is available to take a grade is obtained. This value multiplied by 100 gives the values for the "Grade, per cent" column in the accompanying table. This method, using however a mechanical efficiency of 90 per cent gives an acceleration of from 2.8 to 3.2 ft. per sec. per sec. for passenger cars.

Question:—I wish to know where I could obtain the following information: Initial boiling-point, end-point and range of distillation of some of the substitute gasoline products now offered on the market. Can you describe any method whereby I can derive the British thermal units and the horsepower of any of these products from their gravity and distillation?

Answer:—The best source of information for distillation data for various gasoline substitutes is the Bureau of Mines in the City of Washington. It may be said that the volatility of most of these fuels does not differ radically from that of commercial gasoline. Some of them containing benzol will show larger percentages, boiling around 200 deg. fahr., but this may have little effect on the end-point. There is no direct relation between the British thermal units or horsepower and gravity or distillation range, except for known classes of hydrocarbons. In a series of articles published in the *Automobile Engineer* from February through August, 1921, Ricardo has shown that the horsepower obtainable from a given engine is very nearly the same for a wide range of fuels. There is thus no definite relation between the British thermal units per pound and the horsepower of an engine, though there is of course a relation between the British thermal units and the horsepower hours per pound of fuel.

As for the relation between British thermal units or the horsepower and the gravity or volatility, if one confines his attention to a single hydrocarbon series such as the paraffin series C_nH_{2n-2}, there is a definite relation between the number of British thermal units and either the volatility or the gravity which can be found in the various tables, such as those of Landolt and Börnstein. The same is true of the other series, but not of their indiscriminate mixtures such as occur in some of the fuels that are now to be found on the market.

OBITUARIES

ARTHUR BENJAMIN BROWNE, of Cambridge, Mass., died at the age of 55 years in his native city on Sept. 16, 1922, from cancer of the larynx, after an illness of 22 months. Major Browne was born, Nov. 29, 1866. He was graduated from the Cambridge High School in 1883, and then studied chemical and mining engineering for 5 years under W. French Smith, of the Massachusetts Institute of Technology. From 1889 to 1897 he pursued a general practice as a mining and metallurgical engineer at Boston. During this time he discovered and patented the original electrolytic process for the manufacture of white lead, which received international recognition as a noteworthy contribution to the art. From 1897 to 1908, Major Browne followed a general engineering practice in Montana and Oregon, which led to specialization in internal-combustion engineering as applied to mining and railroading. For 3 years of this time he was engaged as chief engineer in building the Central Railway of Oregon, which he afterward operated as its superintendent.

Following a prolonged illness in 1911, he spent 2 years in study and development work relating to gasoline and oil engines, giving particular attention to the constant-pressure or Brayton type. The years 1913 to 1917 inclusive were spent in research and development in the field of carburetion. During this period he lived in Branford, Conn., and acted as consulting engineer for the Malleable Iron Fittings Co. of that place. In 1915 he wrote an engineering treatise bearing the title, Handbook of Carburetion, which is one of few authoritative books on the subject treated.

Shortly following the entry of this Country into the World War, Mr. Browne was awarded a captain's commission in the United States Army and assigned to engineering duty in the Sanitary Corps. For several months he was in charge of the production, testing and inspection of ambulance and other truck chassis at the General Motors Truck Co.'s plant in Pontiac, Mich. He was later transferred to the City of Washington, where he was promoted to the rank of Major and served as a representative of the Sanitary Corps on the board of engineers charged with the test and final selection of vehicles for use by the Army. Some months later he was made chief of the motor branch, motor and vehicle division of the Department of Purchase, Storage and Traffic.

Upon his discharge from the service in July, 1919, Major Browne accepted the position of chief engineer and director of sales of the A. J. Detlaff Co., Detroit, from which he resigned about a year later. For several months in 1920 he served as consulting engineer of the Trent Process Co., in the City of Washington. Until a few months prior to his death he was engaged in consulting and development work relating to automotive clutches, carbureters and engines.

Major Browne was elected to Member grade in the Society, May 26, 1913. He served as a member of its Standards Committee and contributed several valuable papers.

WILLIAM T. MURPHY, president and general manager of the Standard Machinery Co., Auburn, R. I., died, July 9, 1922, aged 40 years. He was born, June 1, 1882, at Providence, R. I. In June, 1903, he was graduated from Brown University with the degree of mechanical engineer.

Mr. Murphy was employed as a machinist by the Taft-Peirce Mfg. Co., Woonsocket, R. I., from 1903 to 1904, and then by the Gorham Mfg. Co. in Providence as a designer of special shop equipment until 1905, when he became a tool designer in the automobile department of the American Locomotive Co. in Providence and continued this work until 1907. From 1904 to 1907 Mr. Murphy was also connected with the Rhode Island School of Design as a teacher of evening classes in machine design.

His connection with the Standard Machinery Co. dates from 1907. His specialties were anti-friction bearings and sheet-metal machinery, and he designed numerous ball and roller bearings for automobile, truck and aircraft usage, as well as having also designed and built many machines for producing special automobile parts. Mr. Murphy was elected to Member grade in the Society, April 22, 1918.

Current Standardization Work

DURING the months of September and October 15 meetings of Divisions and Subdivisions of the Standards Committee were held at which progress was made on many important subjects assigned to the Divisions and several definite recommendations were approved for consideration at the Standards Committee Meeting on Jan. 9. The complete recommendations of the various Divisions will be published in the December issue of THE JOURNAL under the usual title of Reports of Divisions to the Standards Committee. Several Division meetings will be held early in November, the dates of these being as follows:

Agricultural Power Equipment Division	Nov. 7	Chicago
Engine Division	Nov. 6	Chicago
Electric Vehicle Division	Nov. 3	Detroit
Passenger-Car Division	Nov. 6	Chicago
Passenger-Car Body Division	Nov. 2	Detroit
Stationary-Engine Division	Nov. 7	Chicago

At the meeting of the Storage Battery Division held on Sept. 15 in New York City final action was taken on recommendations for monobloc containers and isolated electric lighting-plant battery ratings. The Lighting Division Subdivision on Bases, Sockets and Connectors held a meeting on Sept. 19 in Cleveland, reporting at the meeting of the Lighting Division in the same city on the following day. The Truck Division met on Sept. 22 in Detroit, discussion centering on the subjects of three-joint propeller-shafts, power take-off shaft-ends, motor-truck cabs and motor-truck ratings.

The Electrical Equipment Subdivision on Magnet Wire met in New York City on Sept. 22 at which a tentative recommendation was adopted which was considered at the Electrical Equipment Division meeting on Oct. 2 in Rochester. At this meeting reports were also submitted on flexible conduit, magneto mountings, generator and starting-motor mountings and spark-plugs. Detailed accounts of these reports will be found under the various subject-headings in this article. The Frames Division held a meeting on Sept. 25 in Detroit at which definite recommendations for frame and running-board brackets were approved. The Lubricants Division members met on Sept. 26 in Cleveland, the tentative crankcase lubricating oil specifications that were discussed at the Summer Meeting of the Society being reviewed and revised in part. The Motorboat Division held a meeting in New York City on Sept. 28 at which Subdivision reports were submitted covering motorboat couplings, exhaust-manifold connections, tachometer drives and stuffing-boxes. Many subjects were suggested for standardization in connection with boat hardware which will be taken up by the Division in the near future.

The Axle and Wheels and the Ball and Roller Bearings Divisions held a joint meeting in Detroit on Oct. 4, this meeting being followed by separate Division meetings at which subjects of special interest to each Division were discussed. The principal subject for joint discussion was the work of the Subdivision on Passenger-Car Front-Axle Hubs and Differentials. The Iron and Steel Subdivision on Sheet Steel held a meeting in Detroit on Oct. 7. The Springs Division members met in Detroit on Oct. 24, several reports being submitted by the Subdivisions appointed at the previous meeting.

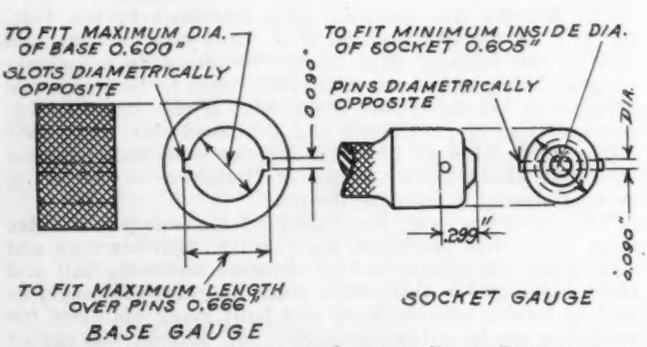


FIG. 1—SUGGESTED GAGES FOR CHECKING BASES, PLUGS AND CONNECTORS

The Iron and Steel Division members met on Oct. 30 in New York City. The Parts and Fittings Division members met in Detroit on Oct. 30, a Subdivision report on Felt Specifications being submitted in the afternoon as a result of a Subdivision meeting held during the morning of the same day. The Transmission Division also held a meeting in Detroit on Oct. 31.

BASES, SOCKETS AND CONNECTORS

The Subdivision on Bases, Sockets and Connectors was appointed to review the present standards for bases, sockets and connectors, pp. B4 and B5 of the S.A.E. HANDBOOK, and to recommend such revisions as are necessary to make them satisfactory for truck practice. At the meeting of the Subdivision on Sept. 19 in Cleveland, it was considered, however, that the present standards are so firmly established in practice that it would be unwise to recommend any changes. A proposal to change the limits on the lamp base, which are now 0.592 and 0.600 in., so as to reduce the allowable tolerance, was considered impossible as the wall is so thin that distortion occurs when the hot glass bulb is inserted. Manufacturers hold the lamp base to within 0.003 in., but the additional 0.005-in. tolerance is necessary in order that the finished lamp may come within production requirements.

It was thought desirable to add an additional dimension to facilitate checking, this dimension to be the distance from the end of the pin to the end of the base, including the solder; to eliminate the notes stating that the pins and slots shall not vary from the center-lines by more than 0.003 in.; and to specify the dimensions of a ring and plug gage, for use in checking the location of the pins for both the base and the plug and in checking the slots in the connector, in accordance with the dimensions specified in Fig. 1.

To provide for manufacturing tolerances, it was also recommended that the slots in the connector tube should be changed from a fixed dimension of 3/32 in. to limits of 0.094 and 0.104 in., and that the following notes referring to the focusing of sockets should be inserted:

> All sockets in which focusing-type electric incandescent lamps are used shall be provided with some means for holding the bulb base firmly throughout its entire length in the socket-shell, because of the difficulty in maintaining the tolerances specified for the base and socket and to eliminate variations in the alignment of the light source.

As the sockets specified in the present standards are of the old spring-plunger type that has given trouble because of poor electrical contact, it was recommended that the design of the socket be changed to one with the spring on the outside of the plunger, inserting the following note:

> It is recommended that all sliding contacts and current-carrying springs be eliminated and that plungers with springs on the outside be used so that the current will pass through the plungers and not through the springs.

As the present standards do not cover the metal type of plug and cap, it was decided that the Subdivision would extend the present standards to cover this type so as to permit interchangeability. Such standardization was considered important as several automobile companies are using armored lighting-cable which cannot be used with the composition cap specified in the present standards.

The Subdivision reported progress at the meeting of the Lighting Division on Sept. 20 and in view of the importance of the work the Subdivision personnel was extended to in-

clude representatives of automobile companies, the complete personnel now being as follows:

C. E. Godley, chairman	Edmunds & James Corporation
Ernest Wooler	Cleveland Automobile Co.
T. I. Walker	Providence Base Works
J. C. Stearns	Culver-Stearns Mfg. Co.
A. K. Brumbaugh	Autocar Co.
G. A. Walters	Chicago Electric Mfg. Co.
J. T. Caldwell	National Lamp Works

The elimination of the insulated-return system was discussed at the Division meeting, but as this system is used to some extent on motor trucks, it was retained in the standard. It was thought that most of the trouble experienced in motor-truck practice is due to the unsubstantial construction of lighting fittings, the heavy vibration causing them to fail. It was suggested that a series of large and more substantial fittings should be standardized.

Body Nomenclature

An editorial recently appeared in one of the automobile magazines criticising the action of the Society in recommending the term "phaeton" instead of "touring car." The following, quoted from a letter written by Geo. J. Mercer as an answer to the criticism, will be of value to anyone interested in considering the justification of the term "phaeton."

> The writer agrees heartily with that portion of your editorial in which it is stated, "It is usually a mistake to overturn standards, once the public has established them." It was precisely along these lines that the committee worked in recommending a name for each type of body. Due consideration was given to established practice and to the fact that the committee was setting a standard for the future as well as for the present. Consideration was given also to the origin or derivation of the names.
>
> As proof that the committee bore the above considerations in mind, take two of the names selected, "brougham" and "cabriolet." Neither has a derivative sense as used. There are several outstanding features that were characteristic of the type of body known as the brougham in the carriage days and in the early days of the automobile, that are lacking in the automobile body which the committee has selected to bear this name. At the Salon one or two genuine broughams are invariably exhibited. Yet the committee has given this name to the type shown on p. K23 of the S.A.E. Handbook because the general public had seen fit to use the name selected. In addition, it was felt that the older or original type was not suitable, in general, for use as an automobile body type. We agreed, therefore, to pass nomenclature on to successors and in naming both the brougham and the cabriolet, we were actuated solely by the fact *that the public commonly recognized these bodies by the terms mentioned.* The writer wishes to add that he has, until within the past two years, combatted the use of these names for these now established types and has been won over to their use only because he has been guided by the principle of usage.
>
> In naming the bodies at this late date we felt that we would meet with considerable opposition, because in nearly every case we had to choose one of several names, but uptodate your editorial is the only objection voiced on behalf of the automobile public. We have had the commendation and the support of the Automobile Body Builders' Association which has unanimously accepted the nomenclature.
>
> One reason the committee chose the name "phaeton" is that mentioned in your editorial; the term "touring" applies to any type of body because all types are used for touring, but that is not the only reason. This type of body is the logical successor of the family horse-drawn open carriage known as the phaeton. It was so called in the earliest handbooks issued by the Association of Licensed Automobile Manufacturers. In the 1906 Handbook, Packard, Walters and Stevens-Duryea used the name "phaeton," Pierce-Arrow designated the vehicle in question as a "tonneau" and others used simply the series letters or numbers of the respective models without names. Later, as the design was changed and became a flush-side body such as we know to-day, "torpedo" was the common name in Great Britain and was used to some extent in this Country; while in France the use of the name "phaeton" was general. Packard, Franklin, Cadillac, Stevens-Duryea, Cunningham, Pope-Hartford and Thomas used the name "phaeton" in the 1913 Association of Licensed Automobile Manufacturers Handbook. Hudson has been a consistent user of this term.
>
> This certainly warranted the committee in concluding that the name "phaeton" has a logical right from a priority point of view, based solely on automobile custom, to contend with the name "touring car." Furthermore, as a name it has a wonderful background of history. It is associated with the type of carriage that people have owned and used beyond our memory. It is also applicable to the more expensive carriages that were exhibited at the horse shows.
>
> As it was the task of the committee to select one name from the several in use for each of the various types of body, due consideration was given to the claims of all and the decisions were based on the facts outlined herein. We feel that our work has met with general approval.

Engine Rotation

At the meeting of the Motorboat Division members on Sept. 28, the possibility of standardizing the direction of engine rotation was discussed in conjunction with replies received from motorboat engine manufacturers. It was the consensus of opinion that it would be impossible to recommend definitely either right or left-hand rotation, but that definite nomenclature should be approved as to what constitutes right and left-hand engine rotation.

Lamp Glasses

At the last meeting of the Lighting Division it was suggested that the present standard for lamp glasses, p. B6 of the S.A.E. Handbook, should be revised so as to make it more satisfactory for various devices used by lens manufacturers. A semi-circular notch was suggested, but not approved by the lamp manufacturers as it would require redesigning the lamp doors and conflict with servicing the large number of lamps now in use. It was decided, however, to modify the form of the notch and a tentative recommendation was approved which will be tried out by the lamp manufacturers before it is finally approved by the Division.

Crankcase Lubricating Oils

The Lubricants Division submitted a report at the Summer Meeting of the Society covering tentative specifications for crankcase lubricating oils, the report being printed in full on p. 533 of the June issue of The Journal.

Subsequent to tentative approval of this report the Division has carried out a large number of tests on five different kinds of oil. The tests were made by various members of the Division on samples of the oil, identified only by number, which were distributed by the chairman. The results of the tests were discussed at the meeting of the Lubricants Division on Sept. 26 and indicated that the error in testing oils in accordance with the proposed specifications would vary considerably even among those actually connected with the petroleum or the automotive industries. It was therefore felt that the limits should not be too close, but on the other hand should be close enough to result in fairly representative test results. It was generally agreed that the principal purpose of the proposed standard is to provide information to the automotive engineers that will be satisfactory as a purchase specification. The specifications as printed in the June issue of The Journal were further amplified by the

inclusion of general descriptive grade names in addition to the specification numbers. This action was taken in view of the general opinion that the specifications would be more widely used if both methods of identifying the different oils were recommended. Other changes were made in the various values as published in June, the revised tentative specifications being given in the accompanying table.

In view of the work of the Governmental Interdepartmental Committee and the American Petroleum Institute, it was considered advisable at the meeting to arrange for a joint session of these organizations and the Lubricants Division in order that a common specification might be formulated which would meet with general approval.

CRANKCASE LUBRICATING OILS

General. — These specifications cover grades of petroleum oil for the lubrication of internal-combustion engines, except aircraft, and are not recommended for the lubrication of turbines.

Compounded lubricating oils containing products other than those derived from petroleum are not dealt with in these specifications.

Spec. No.[1]	Supplementary Grade Names	Flash-Point, Deg. Fahr., Min.	Fire-Point, Deg. Fahr., Min.	Viscosity, Saybolt Sec.				Color[2] (NPA)	Pour Test, Deg. Fahr., Max.	Conradson Carbon Residue, per cent Max.	Corrosion Test
				100 Deg. Fahr.		210 Deg. Fahr.					
				Min.	Max.	Min.	Max.				
20	Light	325	365	180	220	42	....	5	35	0.20	
020	Light	325	365	180	220	42	....	5	10	0.20	
30	Medium	335	380	270	330	44	....	5	40	0.30	
030	Medium	335	380	270	330	44	....	5	10	0.30	Required for all grades
40	Heavy	345	390	360	440	46	....	5	45	0.40	
50	Extra Heavy	355	400	450	575	50	....	6	50	0.60	
60	xx Heavy	360	....	....	....	55	65	....	55	0.80	
80	xxx Heavy	380	....	....	....	75	85	....	55	1.50	
95	xxxx Heavy	390	....	....	....	90	100	....	55	1.75	
115	xxxxx Heavy	400	....	....	....	110	120	....	60	2.00	

[1] For Specifications Nos. 20 to 50, inclusive, the numbers indicate the first two figures of the average Saybolt viscosity in seconds at 100 deg. fahr. for the grades indicated. The first cipher in Specifications Nos. 020 and 030 indicates that the pour-test value of these two grades is 10 deg. fahr. The numbers for Specifications Nos. 60 to 115, inclusive, indicate the average Saybolt viscosity in seconds at 210 deg. fahr.

[2] Darkest color allowed for a mixture of 50 per cent oil and 50 per cent kerosene.

Corrosion Test.—The following corrosion test shall not cause discoloration of copper strip. Place a clean piece of mechanically polished pure strip-copper about ½ in. wide and 3 in. long, and 10 cc. of the oil to be tested, in a clean test-tube. Close the tube with a vented stopper and hold for 3 hr. at 212 deg. fahr. Rinse the copper strip with sulphur-free acetone and compare it with a similar strip of freshly polished copper.

MOTORBOAT NOMENCLATURE

It was brought out at the last meeting of the Motorboat Division that the establishment of nomenclature for engine and motorboat parts is of vital importance to the motorboat industry and that the present automobile engine nomenclature now published in the S.A.E. HANDBOOK could be used to advantage as a basis for such nomenclature. A Subdivision was appointed, consisting of Leonard Octman, Jr., W. J. Deed and Irwin Chase, to initiate the work, it being understood that the personnel will be added to as the work shall require. The Society was asked to obtain catalogs and parts-lists from engine and motorboat builders for use in the Subdivision work.

MONOBLOC CONTAINERS

At the Storage Battery Division meeting on Sept. 15 the tentative specifications for monobloc containers were reviewed and several changes made therein as a result of comments which had been submitted. The specification was approved for adoption as S.A.E. Recommended Practice in the following form:

MONOBLOC CONTAINERS FOR STARTING AND LIGHTING BATTERIES

Height. — Containers shall be made in two heights only; namely, the B height for plates approximately 4¾ in. high, and the C height for plates approximately 5¼ in. high.

Overall Height.—The height from the outside of the bottom of the case to the top of the handle shall not exceed 9¼ in. for B containers, or 9⅞ in. for C containers.

Height from Top of Ribs to Top of Container.—These heights shall be:

B containers 6½ in. } Plus 0
C containers 7 in. } Minus 1/16 in.

Maximum variation between different compartments, 3/32 in.

Inside Width of Compartments.—5 31/32 in., plus or minus 1/32 in.

Inside Lengths of Compartments.—(a) 6-Compartment Containers

B Containers	C Containers
S-3-B 1 5/16 in.	S-4-C 1 1/2 in.
S-4-B 1 1/2 in.	S-5-C 1 11/16 in.
S-5-B 1 11/16 in.	

(b) 3-Compartment Containers

B Containers	C Containers
S- 7-B 2 1/16 in.	S- 8-C 2 3/8 in.
S- 8-B 2 3/8 in.	S-10-C 2 13/16 in.
S- 9-B 2 7/16 in.	S-13-C 3 1/4 in.
S-10-B 2 13/16 in.	S-14-C 3 5/16 in.
S-16-B 3 11/16 in.	S-16-C 3 11/16 in.
S-18-B 3 15/16 in.	
S-19-B 4 1/8 in.	

(c) Tolerances of plus 0 and minus 1/32 in. shall apply to container lengths of 3 5/16 in. or less and tolerances of plus 1/64 in. and minus 1/32 in. to lengths of over 3 5/16 in.

Partitions Between Compartments.—The thickness of the partitions between compartments shall be 3/16 in. minimum and ¼ in. maximum.

Overall Width.—The overall width shall not exceed 7½ in.

GENERATOR AND STARTING-MOTOR MOUNTINGS

At the meeting of the Electrical Equipment Division on Oct. 2, T. L. Lee, as chairman of the Subdivision on Gen-

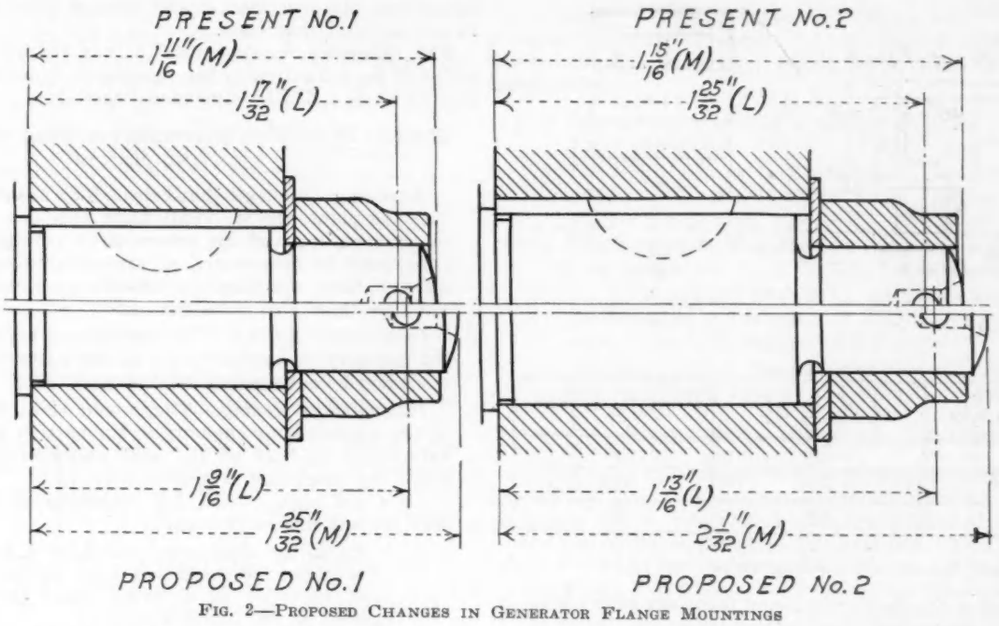


FIG. 2—PROPOSED CHANGES IN GENERATOR FLANGE MOUNTINGS

erator and Starting-Motor Mountings, submitted several recommendations relating to the present standards. As these recommendations had been submitted to generator and starting-motor manufacturers and users and had met with general approval, the Electrical Equipment Division approved them as given herewith.

GENERATOR FLANGE MOUNTINGS

As considerable trouble has been experienced with the present S.A.E. Standard Nos. 1 and 2 generator flange shaft extensions because there is insufficient room for a washer between the nut and the pinion and because of the location of the cotter-pin, the Subdivision recommends that dimension *L*, Fig. 2, be increased by 1/32 in. and dimension *M* by 3/32 in. The revised dimensions allow sufficient room for a washer to be used and locate the cotter-pin hole so that it will not break into the shaft center.

To provide engine builders with all information necessary for making provision for the mounting of S.A.E. Standard flanges, the Subdivision also recommends that

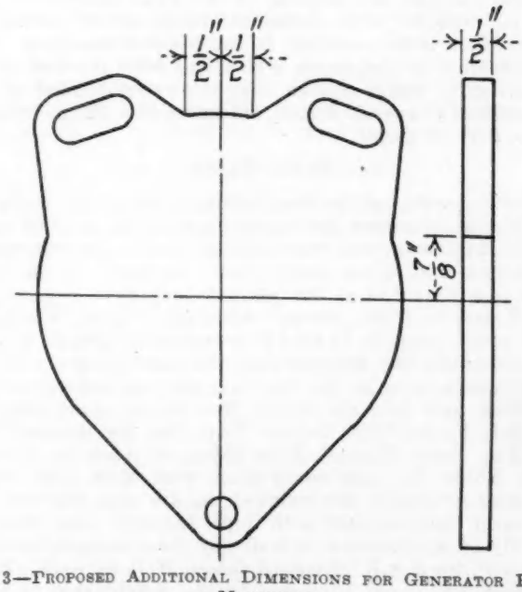


FIG. 3—PROPOSED ADDITIONAL DIMENSIONS FOR GENERATOR FLANGE MOUNTINGS

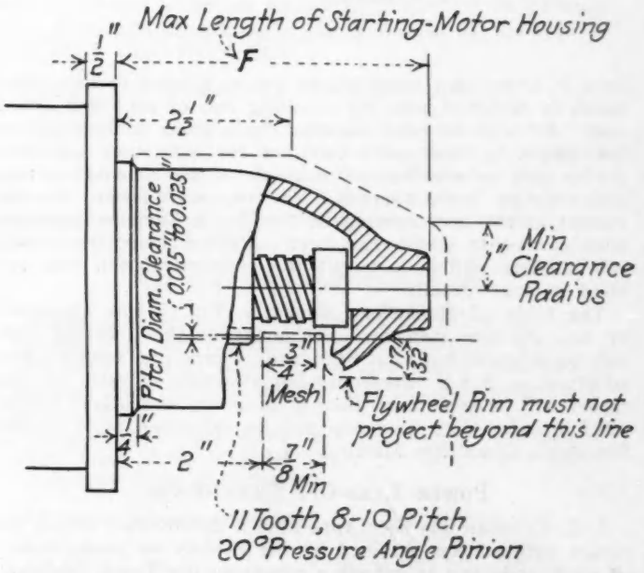


FIG. 4—PROPOSED MOUNTING DIMENSIONS FOR STARTING-MOTOR FLANGE

Dimension *F* Is 4 7/16 In. Maximum for the 10, 11, 12 and 13-Tooth Small Hollow Shift and 4¾ In. for the 13-Tooth Large Hollow Shift

the additional contour dimensions shown in Fig. 3 be included in the present standard and that the thickness of the flange be specified as ½ in.

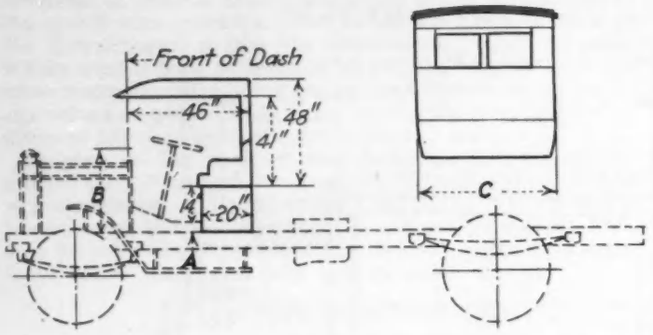
STARTING-MOTOR FLANGE MOUNTINGS

In view of the demand for an S.A.E. Standard for Starting-Motor Flange Mountings for outboard installations only, the Subdivision recommends the adoption of the mounting dimensions given in Fig. 4.

The Subdivision also recommends the adoption of a flange thickness of ½ in. for the present S.A.E. Standard for Starting-Motor Flange Mountings which is printed on p. B19 of the S.A.E. HANDBOOK.

MOTOR-TRUCK CABS

Early last spring it was decided to undertake the standardization of motor-truck cab mounting-dimensions as it was considered important to make possible the production of stock



Truck Capacity, Tons	1, 1½, 2, 2½	3, 3½, 4, 5
A	3½	4
B[1]	35	39
C	48	56

[1] Minimum for windshield lower edge

Where cab sides, doors or side curtains are used, care should be taken that the driver's vision is interfered with as little as possible

FIG. 5—MOUNTING DIMENSIONS OF MOTOR-TRUCK CABS

cabs in order that users might not be obliged to have cabs specially designed with the resulting loss of time and added cost. Information was obtained from truck builders as to the mounting dimensions required for cabs used on their trucks, this information being tabulated and a tentative recommendation based thereon. The recommendation was reviewed at the last meeting of the Truck Division together with comments which had been obtained from truck and cab builders and as a result the recommendation was revised in some details.

The table of dimensions shown in Fig. 5 was approved by the Division members subsequent to this meeting and will be submitted at the Standards Committee meeting for adoption as S.A.E. Recommended Practice. Copies of the report will be sent to all Society members and others interested in order that comments may be submitted prior to the Standards Committee Meeting.

POWER TAKE-OFF SHAFT-ENDS

J. R. Coleman has been appointed a Subdivision of one to review information obtained by the Society on power take-off shaft-ends and to submit a report to the Truck Division for consideration.

RATINGS OF STORAGE BATTERIES FOR ISOLATED ELECTRIC LIGHTING-PLANTS

The history of the Standards Committee work in endeavoring to establish ratings for storage batteries for isolated electric lighting-plants was reviewed at the Sept. 15 meeting of the Storage Battery Division, with particular reference to the action of the Isolated Electric Lighting-Plant Division in recommending at a Division meeting held on July 28, 1921, a standard rating of capacity expressed in terms of watt-hours based on a continuous 8-hr. discharge test, which recommendation did not receive general approval as determined by a letter ballot of storage battery and lighting-plant manufacturers.

In view of the fact that at the present time several methods of rating storage batteries are used, it was considered advisable to recommend that either the intermittent or the continuous ratings should be used in accordance with the recommendation of the Storage Battery Subdivision of the Electric Equipment Division made at a meeting on May 28, 1919, in Indianapolis, the present Storage Battery Division being the successor of the Storage Battery Subdivision in existence at that time.

The following recommendation was therefore formulated and will be submitted to the Standards Committee for adoption as an S.A.E. Recommended Practice:

RATINGS OF STORAGE BATTERIES FOR ISOLATED ELECTRIC LIGHTING-PLANTS

Lead-acid storage batteries for isolated electric lighting-plant service shall have two ratings, a continuous rating and an intermittent rating. The ratings shall be determined at an initial temperature of 80 deg. fahr. and shall be based on a final voltage of not less than 1.75 volts per cell.

Continuous Rating.—The continuous rating shall be the capacity in ampere-hours of the battery when it is discharged continuously at the 8-hr. rate.

Intermittent Rating.—The intermittent rating shall be the ampere-hour capacity of the battery at a current rate equal to 1/24 of the said ampere-hour capacity when the discharge is spread over 72 hr.

To avoid night work, the following discharge and rest periods are suggested:

> An initial discharge period of 4 hr., followed by a 16-hr. rest; then two 8-hr. discharge periods, each followed by a 16-hr. rest; the final discharge period being 4 hr.

The short periods at the beginning and at the end permit the test to begin at noon of the first day and end at noon of the fourth day.

Except for the final voltage per cell the ratings apply to nickel-iron batteries as well as lead-acid batteries.

Battery manufacturers should specify both ratings in their catalogs using the following form:

S.A.E. Ampere-Hour Capacity Ratings: 100-140

The first number represents the capacity based on the continuous rating and the second number the capacity based on the intermittent rating.

In making this recommendation it was understood that if one of the two ratings proposed is eliminated by subsequent Standards Committee or Society action, the Division recommendation shall be referred back as it is considered that the situation can be covered adequately only by specifying both the continuous and intermittent ratings.

RUNNING-BOARD BRACKETS

The present S.A.E. Recommended Practice for Running-Board Brackets printed on p. H23 of the S.A.E. HANDBOOK was reviewed at the meeting of the Frames Division members on Sept. 25 with special reference to the thickness of stock, which is not specified in the present standard. After consideration of comments which had been received in this connection it was proposed that the recommended practice be amplified to specify a stock thickness of 5/32 in. (0.156 in. or No. 9 U. S. gage).

SPARK-PLUGS

At the meeting of the Spark-Plug Subdivision on Aug. 17 the present standard for spark-plug shells, p. A10 of the S.A.E. HANDBOOK, was reviewed in detail. In reference to the thread size, it was stated that two-thirds of the spark-plugs manufactured at the present time have a ½-in. pipe thread, but that the general objection to pipe threads for spark-plugs seems to be that it is extremely difficult to maintain accurately the distance that the spark-plug screws into the cylinder, owing to the fact that the tap and dies are not run down to a constant depth; that metric spark-plugs are used only by the Wills Sainte Claire Co., the Hudson Motor Car Co., Essex Motors, Rolls-Royce of America, Inc. and White Motor Co.; that spark-plugs with ¾-in. pipe threads are being gradually discontinued, as are also the few 5/16 and ⅜-in. spark-plugs with pipe threads; and that the majority of spark-plugs *exclusive of the sizes mentioned* are made with the S.A.E. Standard thread of ⅞ in. —18. It was therefore considered advisable by the Subdivision to retain

the present standard thread as well as all dimensions affecting that portion of the spark-plug shell below the gasket.

In discussing the width across the flats of the hexagon or the "hex diameter" as it is generally called, it was brought out that 20 per cent of the spark-plugs with a ⅞-in. —18 thread have a ⅞-in. hex diameter, 20 per cent have a 1⅛-in. hex diameter and 60 per cent have a 15/16-in. hex diameter. Although there are two or three large users of the ⅞-in. hex spark-plugs, it was considered inadvisable to continue to recommend the use of this size because it does not leave sufficient wall to stand up under wrench strain, especially in two-piece spark-plug construction. It was recognized that it is possible to design two-piece spark-plugs for a ⅞-in. hex but as it would necessitate special inside parts that would not be used on the standard one-piece spark-plug, expensive tool-changes would be required. It would be, however, possible for machinery now used in making ⅞-in.-hex spark-plugs to be reset so as to make the 15/16-in.-hex without additional tooling expense.

The Subdivision considered it advisable to recommend dimensions that would permit interchangeability of different makes of spark-plug having the same type of terminal, and therefore proposed for adoption the following terminal dimensions:

Threaded Type.—The terminal thread shall be No. 8-32 (0.164-in. diameter) S.A.E. Coarse Thread.

Ball Type.—For the ball type of spark-plug terminal the ball diameter shall be 13/32 in.

Slip Type.—For the slip type of spark plug terminal the ball diameter shall be 13/32 in. and shall have a groove width of 0.043 in. plus 0.006 minus 0.000 and a groove diameter of 0.203 in. plus 0.015 minus 0.000.

Post Type.—For the post type of spark-plug terminal the large diameter of the taper post shall be 0.250 in. plus 0.000 and minus 0.003 and the neck diameter 0.220 in. plus 0.000 and minus 0.008.

At the meeting of the Electrical Equipment Division on Oct. 2, the Division approved the report of the Subdivision after careful consideration of the problems involved. It was recognized that the proposed revisions could not be adopted in practice by certain automobile companies, but it was thought that the revised standards would be adopted at some future time by such companies simultaneously with other changes in engine design.

Starting-Motor Pinions

The Electrical Equipment Division has recommended that the present S.A.E. Standard for Starting-Motor Pinions be revised to read:

Starting motors shall be equipped with an 8-10-pitch, 11-tooth, 20-deg-pressure-angle pinion. The starting motor shall be installed so that the pitch-circle about which the teeth on the pinion are generated will be separated from 0.015 to 0.025 in. from the pitch-circle about which the teeth on the flywheel are generated.

Tail-Lamp Illumination

The establishing of definite specifications for tail-lamp illumination was discussed at the Sept. 20 meeting of the Lighting Division. Tests which were carried out by Division members Godley, Kenyon and Porter were described, it being stated that the results indicate that the following are the principal factors that should be included in a definite specification:

(1) The tail-lamp should be located over the center of the license-plate

(2) The light opening in the tail-lamp should be large enough to illuminate directly the entire length of longest plates used

(3) The candlepower of the incandescent lamp should be specified

(4) The license-plate should be set at an angle with the direction of the beam of a light so as to secure more direct illumination of the plate

(5) The specification should cover the worst combination of letters, figures and colors

Three-Joint Propeller-Shafts

At the time the present S.A.E. Recommended Practice for three-joint propeller-shafts was voted upon by the Society members, one or two criticisms were submitted in reference to the use of the square type of shaft-end. This matter was reviewed at the last meeting of the Truck Division and, as it was felt that it is preferable to use either the taper or spline type of shaft-end, the Division recommended that the present square type be omitted. It was thought that in case a square type of shaft-end is desired for use at this point, the present S.A.E. Standard square-shaft fitting should be used as a basis for the shaft-end design.

Transmission Lubricating Oils

At the meeting of the Lubricants Division on Sept. 26, the Subdivision on Transmission Lubricants submitted a report covering transmission lubricating-oil specifications that was approved with slight revisions for adoption as S.A.E. Recommended Practice. The revised report is given in the accompanying table.

TRANSMISSION LUBRICATING OILS

These specifications cover grades of petroleum oil suitable for the lubrication of transmission gears, differential gears, worm drives and roller and ball bearings used in connection with such motor-vehicle equipment.

Compounded lubricating oils containing products other than those derived from petroleum are not covered by these specifications.

Specification No.	Grade	Flash-Point, Deg. Fahr.,[4] Min.	Viscosity at 210 Deg. Fahr., Saybolt Sec.		Pour Test, Deg. Fahr., Max. (American Society for Testing Materials Method)
			Min.	Max.	
110	Winter	350	100	120	10
160	Summer	450	150	170	45

[4]American Society for Testing Materials, Cleveland Open Cup.

Activities of the Sections

Secretaries of the Sections

BUFFALO SECTION
A. J. Fitzgibbons, 168 Claremont Avenue, Buffalo
CLEVELAND SECTION
E. W. Weaver, 5103 Euclid Avenue, Cleveland
DAYTON SECTION
R. B. May, Dayton Engineering Laboratories, Dayton
DETROIT SECTION
Thomas J. Litle, Jr., 733 Seyburn Avenue, Detroit
Mrs. B. Brede, Assistant Secretary, 416 Capitol Theater Building, Detroit
INDIANA SECTION
B. F. Kelly, Weidely Motors Co., Indianapolis
METROPOLITAN SECTION
R. E. Plimpton, 129 East 45th Street, New York City
MID-WEST SECTION
H. O. K. Meister, Hyatt Roller Bearing Co., 2715 South Michigan Avenue, Chicago
MINNEAPOLIS SECTION
Phil N. Overman, 10 South 10th Street, Minneapolis
NEW ENGLAND SECTION
V. A. Nielsen, 701 Beacon Street, Boston
PENNSYLVANIA SECTION
Edward L. Clark, Hunting Park and Rising Sun Avenues, Philadelphia
WASHINGTON SECTION
Benjamin R. Newcomb, 211 Victor Building, City of Washington

A GLANCE through the following reports of Sections meetings activity will convince any member that the local meetings of the Society are constantly rising to a higher plane. The papers and speakers selected reflect the careful planning of the Sections officers in their effort to make Sections meetings attractive in themselves without requiring the misdirected appeal for loyalty and support to build audiences. If you are taking a passive interest in Sections meetings, particularly those in your own district, you are overlooking an opportunity to secure advanced engineering thought that has a tangible value in your work.

There is a mistaken idea in some quarters that Sections meetings are inferior in value and interest to the national Society meetings. Nothing could be farther from the truth. It is not unusual to find Section papers that surpass those given at the Annual and Summer meetings. Discussion is very thorough at meetings of the Sections because of the longer time available for this phase of the program. The author has time to go into his subject in a very comprehensive way. There is an atmosphere of informality in Sections meetings which approaches that of the factory conference. Regular attendance soon enables one to maintain a personal contact with the engineers and production men in the industry that leads to the forming of intimate friendships which cannot easily be acquired in any other way. Study the following paragraphs and attend the meetings that concern you and your branch of the industry.

Schedule of Sections Meetings

NOVEMBER

1—MINNEAPOLIS SECTION — Manufacture of Gray Iron Pistons—George G. Bouthinon

9—INDIANA SECTION—The Relation of the Engineering and Service Problems — Fred E. Moskovics

16—METROPOLITAN SECTION — Regulations Governing the Use of Highways—A. L. McMurtry

17—MID-WEST SECTION—BUSES—G. A. Green

17—CLEVELAND SECTION—Gliders—Georg Madelung

17—BUFFALO SECTION — Air-Cooled Engines — C. P. Grimes

23—PENNSYLVANIA SECTION — Leaf Springs — H. B. Winchell

24—NEW ENGLAND SECTION — Automotive Electric Service—M. B. Speer

24—DETROIT SECTION

PHOTOGRAPHS OF SECTIONS TREASURERS

Photographs of the Treasurers of the Sections are printed on the facing page. This completes the series of photographs of the officers of the Sections as far as it has been possible to secure likenesses from the individuals concerned. The following list gives the names of the Treasurers and their Sections.

Name	*Section*
Otto Burkhardt	Buffalo
J. S. Clapper	Minneapolis
H. E. Figgie	Cleveland
G. Walker Gilmer, Jr.	Pennsylvania
W. P. Kennedy	Metropolitan
Robert F. McCann	Dayton
Nelson B. Nelson	Mid-West
L. B. F. Raycroft	New England
Mark Smith	Indiana
Charles S. Whitney	Detroit
Conrad H. Young	Washington

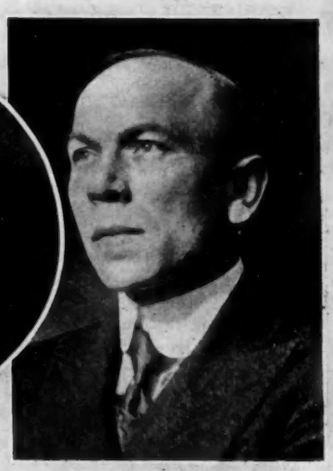
DAYTON SECTION

C. F. Kettering's address before the meeting of the Dayton Section, Oct. 3, was received with enthusiasm and a large audience gathered for the occasion. Mr. Kettering urged a more thorough appreciation of economics on the part of engineers. He discussed the automobile of the future and declared that it would be lighter, more economical and better sprung. Lubrication is one of the most important phases of

G. WALKER GILMER, JR.

ROBERT F. McCANN

J. S. CLAPPER



OTTO BURKHARDT

CHARLES S. WHITNEY

L. B. F. RAYCROFT

W. P. KENNEDY

NELSON B. NELSON

CONRAD H. YOUNG

MARK SMITH

H. E. FIGGIE

automotive engineering requiring study and engines must be designed to handle low-grade fuels.

The Dayton Section will not meet during November. P. S. Tice will address the Section on the Utilization of Low-Grade Fuels in Automotive Engines at its next meeting which is scheduled for Dec. 19.

New England Section

O. J. Rohde's paper on Wheels excited great interest at the October meeting of the New England Section because it revealed the conclusions that have led an established wire wheel company to enter the steel-disc wheel field. Mr. Rohde still considers the wire wheel the best type for passenger-car use, but he finds that the steel-disc wheel possesses the strength and flexibility qualities of the wire wheel at a greatly reduced production cost. Important improvements in wire wheel construction were described by the author, particularly the advancements made in spoke material and design. Methods of testing wire and steel wheels were shown with motion pictures. Mr. Rohde concluded that the wire wheel still possesses greater strength in combination with lighter weight than all other types of passenger-car wheel.

The next meeting of the New England Section will be held in Springfield, Mass., at the Hotel Kimball on the evening of Nov. 24. The meeting will be addressed by M. B. Speer, superintendent of the automotive service department of the Westinghouse Electric & Mfg. Co. Mr. Speer has been engaged in the automotive electrical field for many years and has specialized in the service end of the business. He has had a broad experience instructing repairmen and service-stations on the proper care and repair of automotive electrical apparatus. Electrical service men from all sections of New England are invited to attend this meeting.

Cleveland Section

The Cleveland Section had a very successful meeting, Oct. 20, when George M. Graham gave an excellent talk on the Relation of the Engineer to the Sales Department.

The great public interest displayed in the recent accomplishments of the German sail-planes has led the Cleveland Section to select this interesting means of flight as the topic for its meeting on Nov. 17. Dr. Georg H. Madelung will present the paper of the evening. Dr. Madelung has been in this Country about a year and was previously connected with the Technische Hochschule, Hannover, Germany, where he worked on the design of motorless gliders for the Rhoen contests. His acquaintance with the pilots and engineers who were active in the establishing of the various motorless flight records assures the presentation of an interesting paper. Dr. Madelung was at one time associated with Dr. Junkers and the Albatros Werke but is now a member of the Glenn L. Martin Co.'s engineering staff. The meeting will be held in the rooms of the Cleveland Engineering Societies, Hotel Winton, starting at 8 o'clock.

Detroit Section

The Detroit Section did not meet in October since all of its energy and interest was concentrated on the Production Meeting of the Society held in Detroit, Oct. 26 and 27 and described elsewhere in this issue.

The Section plans to continue the holding of meetings devoted to production matters and there is a possibility of its scheduling two meetings a month in the future, one on design and engineering, the other on production and shop problems. The next meeting of the Detroit Section is scheduled for Nov. 24, but the topic and the speaker have not been announced. All meetings are held at the Detroit Board of Commerce Building and start at 8 o'clock.

The Detroit Section office was moved Nov. 1 to 416 Capitol Theater Building. The new quarters are larger in size and permit the holding of larger committee meetings in the office. The location is central and readily accessible from all sections of the city. A complete library of Society publications is maintained in this office for the use of the members. The Assistant Secretary, Mrs. B. Brede, keeps in close touch with the New York City office and she is able to serve the members in their relations with the Society. Those members seeking employment will find the Detroit Section office of particular value in locating a new position.

Minneapolis Section

Four-wheel brakes were discussed at the October meeting of the Minneapolis Section. H. E. Morton described the features of the brake system bearing his name and outlined the general advantages of four-wheel brakes. He stated that front-wheel brakes are superior to rear-wheel brakes if either type is used singly. The front-wheel brakes effect a momentary transfer of 25 per cent of the weight of the car to the front axle when braking action takes place. Reasons underlying the lessened tendency to skid with brakes on all wheels were given by the author. Although four-wheel brakes stop a car in a much shorter distance than the conventional type, this particular advantage cannot be utilized in heavy traffic since vehicles in the rear of a car so equipped would collide with it if quick stops were made. The big advantage in this instance is the decreased wear and tear on tires and brakes with the four-wheel system which stops the car effectively without putting heavy loads on the brake bands, locking the wheels or sliding the tires.

N. S. Kingsley gave an illustrated talk on the origin of gas and crude oil, the treatment of the crude oil in stills and the refinery methods used to produce motor-fuels and lubricating oils.

The November meeting of the Section will be held at the Manufacturers Club, Nov 1 at 8 o'clock. George G. Bouthinon will present a paper on gray iron pistons.

Pennsylvania Section

The Pennsylvania Section met, Oct. 26 at the Engineers Club, Philadelphia, and discussed the production and design of steel passenger-car bodies. Motion pictures were shown depicting the manufacturing processes employed in the building of all-metal bodies.

Automobile Leaf Springs will be the topic at the meeting of the Pennsylvania Section on Nov. 23 The paper of the evening will be presented by Harold B. Winchell who is associated with William & Harvey Rowland, Inc., Philadelphia, manufacturer of automobile springs. Mr. Winchell has done considerable work on the specifications of springs for passenger-car use and has assisted in the development of a progress system of spring production. He will discuss spring action, the mechanics of spring design and stress the importance of careful selection of spring proportions in the early stages of design rather than experimenting to get a result on a finished car.

Washington Section

R. E. Carlson of the Bureau of Standards addressed the October meeting of the Washington Section. Mr. Carlson explained the operation of the apparatus developed by the Bureau for recording the performance data of an automobile while it is undergoing road test. This apparatus makes it possible to study the effect of fuel volatility changes on car performance under actual operating conditions. The particular apparatus described by Mr. Carlson is the one being used in the Society's Cooperative Fuel Research which is being carried on at the Bureau of Standards at the present time.

The date and program of the November meeting of the Section were not decided at the time THE JOURNAL went to press.

The Washington Section decided at its last meeting to secure the active interest of local automobile dealers and service-men in its meetings. Those who heard R. E. Carlson's talk agreed that it was intensely interesting to servicemen and a committee was appointed to get more of them to attend the future meetings.

Indiana Section

One of the largest audiences ever assembled at an Indiana Section meeting gathered on the evening of Oct. 12 to hear an interesting talk by Thomas Midgley, Jr., of the General Motors Research Corporation. The attendance

reached nearly 300 and the Section officers appreciated the enthusiastic send-off for the coming winter program.

The Indiana Section meets, Nov. 9 to hear an address by Fred E. Moskovics, vice-president of the Nordyke & Marmon Co. Few men have studied automobile service with greater care than Mr. Moskovics and his experience and recommendations will deserve the attention of all engineers. The topic selected for this meeting is the Relation of Engineering to the Service Problem. Indiana Section meetings are held at the Athenæum, Michigan and New Jersey Streets, Indianapolis. They are preceded by a dinner at 6:30, the meetings starting at 8 o'clock.

Mid-West Section

C. F. Kettering was the guest of the Mid-West Section at a dinner held at the Engineers Club on Oct. 20. Over 100 members and guests were present and shared the enjoyment of Mr. Kettering's customary mixture of good humor and sound logic. He presented a very thorough analysis of the Fundamentals of Engineering.

The Section has been extremely fortunate in securing G. A. Green, formerly vice-president and general manager of the Fifth Avenue Coach Co., to address its November meeting. The excellent paper given by Mr. Green at the Summer Meeting of the Society aroused great interest in Chicago, where there is an unusual opportunity for the expansion of motorbus transportation. The municipal authorities concerned with Chicago's transportation media will be present at the meeting; many electric railway men are also being invited. This meeting will be held Nov. 17 in the rooms of the Western Society of Engineers, Monadnock Building, starting at 7:30 p. m.

Buffalo Section

L. H. Pomeroy gave a very comprehensive paper on the fundamentals underlying light-weight design at the meeting of the Buffalo Section, Oct. 19. Mr. Pomeroy discussed the conditions determining chassis weight in general as they are related to bearings sizes and their supports. The extent to which bearing loads in engines are a function of inertia rather than gas pressure was studied in the paper. Constructional materials were considered in terms of their specific strength and the effect of forged aluminum connecting-rods on engine performance and efficiency was shown. The paper contained many valuable data for the use of engineers who may be considering the more extensive use of aluminum alloys in automotive engines and chassis.

The next meeting of the Buffalo Section is announced for the evening of Nov. 17. C. P. Grimes, research engineer, H. H. Franklin Mfg. Co., will present a paper on the problems encountered in the design and development of air-cooled automobile engines.

Metropolitan Section

William B. Stout read a paper on the Modern Airplane at the October meeting of the Metropolitan Section and told the members that he believed the commercial airplane of the future will be of the all-metal monoplane type; the inability of wood veneer and linen to withstand severe service in inclement weather will drive them both out of commercial plane structure. The monoplane with internally-trussed wings of thick section lends itself admirably to all-metal construction. Thick wings can be selected for high-lift characteristics and the elimination of struts and cables reduces the head-resistance to a minimum. Mr. Stout exhibited samples of some very stiff rolled duralumin sections that he had developed and described a large all-metal torpedo plane in which these were employed. He believed that relatively low-power engines would be used in the commercial airplanes of the future because of their lower first cost.

The Metropolitan Section will be addressed by A. L. McMurtry at its meeting on Nov. 16. Mr. McMurtry was formerly engineer of the Connecticut State Motor Vehicle Commission and has been largely responsible for the constructive highway legislation program of that State. His experience in highway traffic regulation is very broad and motor-truck operators and fleet owners will find his talk particularly valuable to them. Highway engineers and members of city planning commissions will be invited to attend the meeting and discuss Mr. McMurtry's paper. The meeting will be held at the Automobile Club of America at 8 o'clock, being preceded by the customary dinner at 6:30.

IMPACT TEST

THE impact test reveals the amount of work that must be done on a given specimen to produce rupture or a certain deformation. In the usual tension-test, if the elastic-limit is determined the engineer can safely apply this result in figuring the dimensions of any member of a structure. In the case of the impact test, the results cannot be so applied. The work required to break any given section cannot be calculated from the work required to fracture a test-bar of different dimensions. This does not, however, destroy the value of the impact test. Its comparative value is still vital in passing upon the properties of a given material for certain specific purposes.

The above statements are based upon results obtained from notched test-bars broken as beams, supported, as the case may be, at one or both ends. It may not represent conditions for notched tension-tests, but sufficient data are not yet available to make a definite statement regarding this point.

How then can the results of the impact test be interpreted in engineering practice? This can probably best be answered by the following illustration. No component of a gun, such as the jacket or tube, which has ruptured in service from a cause that cannot definitely be assigned to defective ammunition, excessive power pressure, etc., has shown an impact value exceeding 6 ft-lb. on specimens tested in a small Charpy impact-machine. It has therefore become the practice of Watertown Arsenal to take impact tests from all components of a gun and, regardless of their ordinary tensile properties and the acceptability of the forging as judged from this test, no piece is accepted unless the results on the small Charpy impact-machine show a value of at least 6 ft-lb. To have a factor of safety, no major forging would be accepted unless this result was at least 10 ft-lb. It is believed that for any class of service a careful study of failures, together with material that is satisfactory, will enable the determination of a critical impact-value for a specific service.

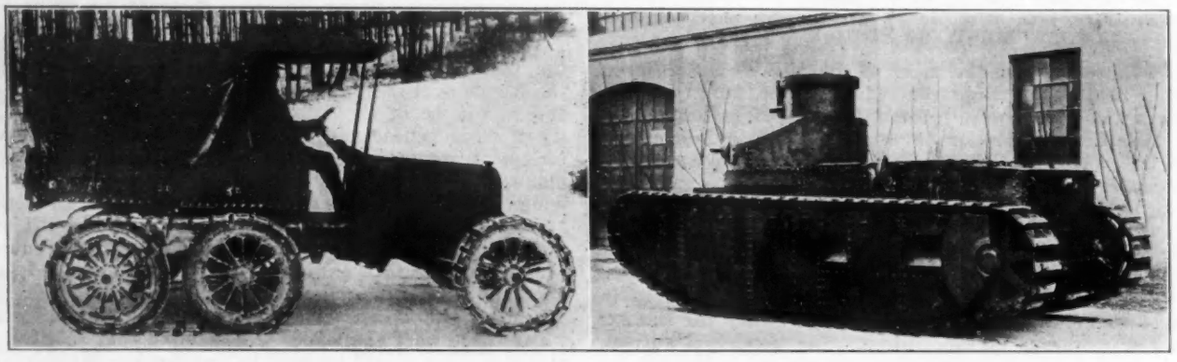
Although not necessarily pertinent to the subject under discussion, it is undoubtedly a fact that many metallurgists and engineers question the value of the impact test on either notched or unnotched specimens. It has been the experience of the laboratories at the Watertown Arsenal that the Charpy impact-test has materially assisted in the diagnosis of many failures of parts in service and has made possible the prevention of their recurrence when applied to the testing of replacement parts. Although no definite relationships exist between the static tensile-properties as conventionally used and the results of the Charpy tests, this in itself cannot condemn the Charpy test or any other form of impact test on either notched or unnotched bars. Many specific cases could be cited where the impact test has been of the utmost practical value, and in view of this condition it is not believed that condemnation of the test is justified.—From a paper by F. C. Langenberg and N. Richardson, presented at meeting of the American Society for Testing Materials.

Aberdeen Proving Ground Visit

THE members of the Society, through the courtesy of Gen. C. C. Williams, chief of ordnance, again enjoyed the privilege of visiting Aberdeen Proving Ground, Md., where a program of test firings and demonstrations of post-war ordnance material was held on Oct. 6. The arrangements for this year's meeting were similar to those made for 1921, and this, without detracting from the interest of the exhibit, served to emphasize the progress made during the year. Thus, firing the 16-in. gun was again the first event, but the new feature, the recently perfected solenoid chronograph used for determining the velocity of the projectile, excited as much comment as the more spectacular shooting of the gun. Similarly, the railroad mount for 14-in. guns and 12-in. howitzers was shown in a design allowing of greater mobility during emplacement than was possible last year. The distinctly new developments shown for the first time were the supersensitive fuzes and the nitrotite containers; the former serve to explode anti-aircraft projectiles by impact against so slight an obstacle as two thicknesses of airplane silk, while the latter protect explosive charges, even when completely immersed in water, against the effects of moisture.

The automotive apparatus circus, which so pleased the Society members last year, was staged once more. Among its novelties this year were the medium A tank described below, a radio-directed whippet tank and the Mark VIII tank with a revolving observation tower operating on the stroboscopic principle. The " amphibious" tank, however, omitted its amusing performance of last year.

In the general run of automotive development in the Ordnance Department there are but few features which, taken singly, are not of interest to the commercial designer and builder of trucks and tractors. For example, the medium A tank, illustrated on this page, has a new type of track suspension, and the construction of the track shoes proper employs novel means to secure more efficient lubrication of the track pins than has been the practice in previous types of track-laying vehicles. Another instance, also illustrated herewith, is the Dodge truck to which has been added a fabric track. The purpose of the track application to a commercial vehicle is to provide increased mobility in military service. It is hoped that all general cargo work for supplying an army in the field will be accomplished in the future by the use of this or a similar type of vehicle. In case this development is ultimately successful, there will be a field for its general use on the farm and for road transport in those sections of the country where there are few or practically no improved roads.



TWO PIECES OF AUTOMOTIVE ORDNANCE MATERIAL THAT WERE SHOWN TO THE MEMBERS ON THEIR VISIT TO THE ABERDEEN PROVING GROUND ON OCT. 6

At the Left Is a Dodge Truck Equipped with a Fabric Track That Is Expected To Provide Increased Mobility in Military Service. The Tank at the Right Is Characterized by a New Type of Track Suspension and the Use of Novel Means in the Construction of the Track Shoes To Secure More Efficient Lubrication of the Track Pins

Other exhibits of interest to the automotive engineer were much mobile artillery material, including the Christie mount for 4.7-in. guns, the dirigible D-4, the Owl bombing airplane and the foreign army trucks, tractors, mounts and tanks on display in the Museum. This array of automotive devices makes an impressive demonstration of the important problems for which the Ordnance Department sees an eventual solution in the simplicity and compactness of the gasoline engine as a powerplant.

WORK OF AMERICAN ENGINEERING STANDARDS COMMITTEE

THE growing interest in standardization on the part of almost every American industry is emphasized by the quarterly report of the activities of the American Engineering Standards Committee. Of the projects that have official status before the committee 20 are concerned with mechanical engineering; 17 are civil engineering projects; 15 are electrical; 3 are automotive; 10 are concerned with transportation; 10 with ferrous metals; 11 with chemical; 5 with non-ferrous metals; 4 with mining; 2 with textiles; 1 with shipbuilding; and 8 projects are of general interest.

Twenty-four standards or safety codes have been approved and 36 are up for approval. The remaining 46 projects represent codes and standards that are in the process of formulation or are now being considered by committees of representatives, designated by the various bodies, industrial, technical and governmental, interested in each particular subject. In this way more than 200 such bodies are officially participating in the work of the American Engineering Standards Committee through their accredited representatives.

A regular interchange of information as to the status of work under way is maintained by the American Engineering Standards Committee with the national standardizing bodies of Austria, Belgium, Canada, Czecho-Slovakia, France, Germany, Great Britain, Holland, Italy, Japan, Norway, Sweden and Switzerland.

Applicants *Qualified*

The following applicants have qualified for admission to the Society between Sept. 9 and Oct. 10, 1922. The various grades of membership are indicated by (M) Member; (A) Associate Member; (J) Junior; (Aff) Affiliate; (S M) Service Member; (F M) Foreign Member; (E S) Enrolled Student.

ANDERSON, IVAN L. (E S) student, University of Utah, *Salt Lake City, Utah,* (mail) 1859 Lake Street.

BARROW, ARTHUR THOMAS (A) general manager, City Motor Works, Ltd., Kent Terrace, *Wellington, New Zealand.*

BASCH, JACOB JUSTIN (J) sales engineer, G & O Mfg. Co., New Haven, Conn., (mail) 828 North Broad Street, *Philadelphia.*

BIDDLE, CHARLES JONATHAN (E S) student, California Institute of Technology, Pasadena, Cal., (mail) 2815 Stuart Street, *Berkeley, Cal.*

BITTERMAN, SIMON (A) president and manager, American Auto Products Co., 240 South Broadway, *Denver, Col.*

BOSCHAN, ALEXANDER S. (E S) student, University of Michigan, Ann Arbor, Mich., (mail) 213 First Street, *Keyport, N. J.*

BURN, WALTER P. (A) sales promotion and advertising manager, Transcontinental Oil Co., 510 Benedum-Trees Building, *Pittsburgh.*

BURT, LEO O. (M) designer, Chevrolet Motor Co., *Detroit,* (mail) 1732 West Bethune Avenue.

CHASE, RUSSELL D. (A) P. O. Box 163, *Summit, Cook County, Ill.*

CHESEBRO, C. M. (M) chief engineer, Detroit Motor Co., *Washington, Pa.*

COBB, CARROL J. (E S) student, Ohio State University, *Columbus, Ohio,* (mail) 45 Crestview Road.

COFFIN, O. W. L. (A) branch manager, White Co., 1715 Preston Avenue, *Houston, Tex.*

COX, CAPT. M. R. (S M) U. S. Army, *Fort Sill, Okla.*

CROSS, CHARLES H. (E S) student, Ohio State University, Columbus, Ohio, (mail) 3210 Arthington Street, *Chicago.*

DAVISON, WALTER W. (M) factory manager, Wire Wheel Corporation of America, Inc., 1700 Elmwood Avenue, *Buffalo, N. Y.*

FOGELSON, EMIL (M) mechanical engineer, Victor Pagé Motors Corporation, *New York City,* (mail) 1086 Kelly Street.

GOMMEL, D. E. (J) chief engineer, Monroe Automobile Co., *Indianapolis, Ind.,* (mail) 2905 Meredith Street.

GORDON, JOHN RUTHERFORD (F M) body builder, Holdens Motor Body Builders, *Adelaide, Australia.*

GROVES, FRANK A. (A) president and general manager, Stemco Engineering Co., Second and Webb Streets, *Dayton, Ohio.*

HERMANN, JOHN F. (A) assistant engineer, Phelps Light & Power Co., Rock Island, Ill., (mail) 614 Seventh Avenue, *Clinton, Iowa.*

HIRSH, DAVID H. (J) instructor, Greer College of Automotive Engineering, 2024 South Wabash Avenue, *Chicago.*

HORTHY, WILLIAM A. (M) mechanical engineer, tractor works, International Harvester Co., *Chicago,* (mail) 4828 Dorchester Avenue.

HOWER, HARRY S. (M) director of research, Macbeth-Evans Glass Co., *Pittsburgh.*

HUGHES, F. J. (A) technical representative, Tilling-Stevens Motors, Ltd., *Maidstone, England* (mail) 60 Holland Road.

KARLSON, K. EDWIN M. (M) factory manager, Cleveland Automobile Co., *Cleveland.*

KENNEDY, JAMES T. (A) sales department, Goodyear Tire & Rubber Co., 2817 East Grand Boulevard, *Detroit.*

KNOWLTON, H. B. (M) instructor of metallography heat-treating, Central Continuation School, Seventh and Prairie Streets, *Milwaukee.*

KOHR, ROLAND MEREDITH (E S) student, Ohio State University, Columbus, Ohio, (mail) *New Philadelphia, Ohio.*

MENEWISCH, WILLIAM T. (J) draftsman, Fox Motor Car Co., *Philadelphia,* (mail) 1332 South Divinity Place.

MILLER, HENRY J. (J) draftsman, Zeder-Skelton-Breer Engineering Co., Newark, N. J., (mail) 178 Montgomery Street, *Bloomfield, N. J.*

MOHORI, TERUO (A) 13816 Argus Avenue, *Cleveland.*

MORTIMER, BRUCE G. (A) assistant superintendent, Wilson Motor Sales Co., *Toronto, Ont., Can.,* (mail) 1504 Dufferin Street.

PARISH & BINGHAM CORPORATION (Aff) *Cleveland.*
Representatives:
Conrad, F. K., secretary and treasurer.
Fitch, R. G., sales manager.
Harmon, O. B., assistant chief engineer.
McMunn, William N., vice-president.
Maloney, J. E., assistant sales manager.
Morse, Agnes D., president.

PLUMRIDGE, TOM G. (A) consulting engineer, American Technical Society, *Chicago,* (mail) 4455 Ellis Avenue.

ROEMMELE, HOWARD CARL (E S) student, Stevens Institute of Technology, Hoboken, N. J., (mail) 31 Astor Street, *Newark, N. J.*

SALISBURY, C. E. (M) manager, service department, Hupp Motor Car Corporation, *Detroit.*

SCHENCK, R. B. (M) metallurgical engineer, Buick Motor Co., *Flint, Mich.,* (mail) 708 Clifford Street.

SCHNEIDER, HAROLD P. (E S) student, Ohio State University, Columbus, Ohio, (mail) 123 14th Street, *Toledo.*

SCULLY, JAMES N. (J) president, Houdaille Co., *Buffalo,* (mail) 401 Delaware Avenue.

SEILER, PAUL W. (A) president and general manager, Ternstedt Mfg. Co., *Detroit,* (mail) Box 6, West Fort Station.

SKINNER CO., LTD. (Aff) *Gananoque, Canada.*
Representative: Skinner, Frederick J., president and general manager.

SOWER, GEORGE W. (E S) student, Ohio State University, Columbus, Ohio, (mail) 11721 Oakview Avenue, *Cleveland.*

STUTTER, CLINTON L. (J) assistant foreman, J. Linek, 6 Washington Street, *Maspeth, N. Y.*

SUN CO. (Aff) Finance Building, 1428 South Penn. Square, *Philadelphia.*
Representatives:
Buderus, W. H.
Cox, C. P., manager motor oil department
Drysdale, R. L., cutting oil engineer
Eckert, S. B., district sales manager
Pew, J. N., Jr., vice-president

TALL, G. W. (A) sales manager, electric furnace division, Leeds & Northrup Co., 4901 Stenton Avenue, *Philadelphia.*

TEETOR, LOTHAIR (A) sales manager, Indiana Piston Ring Co., *Hagerstown, Ind.*

TETENS, RAYMOND E. (M) assistant chief engineer, Earl Motors, Inc., *Jackson, Mich.*

VOORHEES, R. C. (A) R. F. D. No. 3, *Ypsilanti, Mich.*

WILSON, EDWARD ARTHUR (E S) student, California Institute of Technology, *Pasadena, Cal.,* (mail) 192 South Cypress Street, *Orange, Cal.*

Applicants *for* Membership

The applications for membership received between Sept. 15 and Oct. 16, 1922, are given below. The members of the Society are urged to send any pertinent information with regard to those listed which the Council should have for consideration prior to their election. It is requested that such communications from members be sent promptly.

BACON, DAY H., designer, Ansted Engineering Co., *Connersville, Ind.*

BISHOP, CHARLES ALBERT, designer, Glenn L. Martin Co., *Cleveland.*

BAYLESS, RAY T., chemical engineer and metallurgist, American Society for Steel Treating, *Cleveland.*

BROOKE, HAROLD LEE, in charge of dynamometer experimental department, Maxwell Corporation, *Highland Park, Mich.*

CALLUM, JOHN R., designer and builder of racing cars, J. R. Callum & Co., Inc., *Norfolk, Va.*

CARNEY, ROY C., service superintendent, Federal Motor Truck Co., *Chicago.*

CAUTLEY, RANDOLPH, investigator of costs and special manufacturing problems, Wright Aeronautical Corporation, Paterson, N. J.

CERVENY, FRANK, instructor, Janesville High School, *Janesville, Wis.*

CHERNIACK, NATHAN, assistant transportation engineer, Ward Motor Vehicle Co., *Mount Vernon, N. Y.*

CHRISTENSEN, E. G., sales manager, Carl Pick Co., *West Bend, Ind.*

COLWELL, DONALD L., chief chemist and metallurgist, Stewart Mfg. Corporation, *Chicago.*

COOKE, B. W., president, Cooke Auto School and Coyne Trade and Engineering School, *Chicago.*

D'ANNUNZIO, UGO V., president, Isotta Motors, Inc., *New York City.*

DEAN, KENNETH G., engineer, Lavine Gear Co., *Milwaukee.*

DELANEY, GEORGE A., engineer, Paige-Detroit Motor Car Co., *Detroit.*

DOLL, GUS. P., treasurer and director of sales, Thomas J. Corcoran Lamp Co., Cincinnati.

EHRKE, MALCOLM N., draftsman, Western Automatic Machine Screw Co., *Elyria, Ohio.*

ELLIS, CAPT. CARROLL L., *Aberdeen Proving Ground, Md.*

FERRIS, WALTER, chief engineer, Oilgear Co., *Milwaukee.*

GREENWALD, H. A., engineer, Cadillac Motor Car Co., *Detroit.*

GUBITZ, WERNER, body designer, Locomobile Co., *New York City.*

HAMPSHIRE, GEORGE W., district manager, Colonial Steel Co., *Pittsburgh.*

HELLINGS, S. A., vice-president and sales manager, Stewart Mfg. Co., *Chicago.*

HOPKINS, FRED. J., designer, International Harvester Co. of America, *Chicago.*

HOPKINS, LUTHER H., maintenance foreman, tractor works, International Harvester Co., *Chicago.*

HUNT, SAMUEL J., manager and lubrication engineer, J. D. Streett & Co., *St. Louis.*

IRVIN, W. A., assistant to vice-president, American Sheet & Tin Plate Co., *Pittsburgh.*

JANSON, OLOF, draftsman, Advance-Rumely Co., *Laporte, Ind.*

KERSTEN, A. EDGAR, service manager, Garford Motor Truck Co., *Boston.*

LEISTER, FAYETTE, sales engineer, Fafnir Bearing Co., *New Britain, Conn.*

LILLIBRIDGE, BYRON J., JR., B. F. Sturtevant Co., *Hyde Park, Mass.*

LIU, CALVIN Y., student, University of Michigan, *Ann Arbor, Mich.*

MACGREGOR, B. N., cable sales, Packard Electric Co., *Warren, Ohio.*

MALCHOW, HENRY J., division engineer, Standard Oil Co. (Indiana), *Milwaukee.*

MONK, MARVIN E., assistant sales manager, U. S. Ball Bearing Mfg. Co., *Chicago.*

PICKARD, THOMAS, inspector, Moon Motor Co., *St. Louis.*

ROBERTS, LOUIS L., experimental engineer, C. H. Wills & Co., *Marysville, Mich.*

ROOKS, ALFRED W., maintenance engineer, Chevrolet Motor Co., *Detroit.*

SHIRASAWA, GEN., engineer, Japan Automobile Co., Akasaka, *Tokio, Japan.*

SMECKEL, ERIC F., assistant chief engineer, Walker Axle Co., *Chicago.*

SMITH, HAROLD E., model maker, Remy Electric Co., *Anderson, Ind.*

TAMPLIN, WILLIAM G., special representative, American Sheet & Tin Plate Co., *Pittsburgh.*

THOMSON, ROBERT, chief draftsman, Ainsworth Mfg. Co., *Detroit.*

THORSON, WILBUR R., assistant manager, Central Iowa Electric Co., *McCallsburg, Iowa.*

TIMMES, JOHN A., office manager, Studebaker Corporation of America, *New York City.*

WANG, CHENG FU, draftsman, Glenn L. Martin Co., *Cleveland.*

WARFORD, HENRY W., student, Tri-State College of Engineering, *Angola, Ind.*

WELCHANS, EDWARD, chief engineer, J & B Mfg. Co., *Pittsfield, Mass.*

WHITE, KARL H., aeronautical engineer, Aeromarine Plane & Motor Co., *Keyport, N. J.*

WOOD, GEORGE A., factory manager, Mutual Wheel Co., *Moline, Ill.*

ZOBEL, CARL G. F., laboratory assistant, Bureau of Standards, *City of Washington.*